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TYPES OF MARINE BOILERS
MARINE-BOILER DETAILS
MARINE-BOILER ACCESSORIES
FIRING
ECONOMIC COMBUSTION
MARINE-BOILER FEEDING
MARINE-BOILER MANAGEMENT
MARINE-BOILER REPAIRS
MARINE-BOILER INSPECTION
PROPULSION OF VESSELS
REFRIGERATION

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PREFACE

The International Library of Technology is the outgrowth of a large and increasing demand that has arisen for the Reference Libraries of the International Correspondence Schools on the part of those who are not students of the Schools. As the volumes composing this Library are all printed from the same plates used in printing the Reference Libraries above mentioned, a few words are necessary regarding the scope and purpose of the instruction imparted to the students of—and the class of students taught by—these Schools, in order to afford a clear understanding of their salient and unique features.

The only requirement for admission to any of the courses offered by the International Correspondence Schools, is that the applicant shall be able to read the English language and to write it sufficiently well to make his written answers to the questions asked him intelligible. Each course is complete in itself, and no textbooks are required other than those prepared by the Schools for the particular course selected. The students themselves are from every class, trade, and profession and from every country; they are, almost without exception, busily engaged in some vocation, and can spare but little time for study, and that usually outside of their regular working hours. The information desired is such as can be immediately applied in practice, so that the student may be enabled to exchange his present vocation for a more congenial one, or to rise to a higher level in the one he now pursues. Furthermore, he wishes to obtain a good working knowledge of the subjects treated in the shortest time and in the most direct manner possible.

In meeting these requirements, we have produced a set of books that in many respects, and particularly in the general plan followed, are absolutely unique. In the majority of subjects treated the knowledge of mathematics required is limited to the simplest principles of arithmetic and mensuration, and in no case is any greater knowledge of mathematics needed than the simplest elementary principles of algebra, geometry, and trigonometry, with a thorough, practical acquaintance with the use of the logarithmic table. To effect this result, derivations of rules and formulas are omitted, but thorough and complete instructions are given regarding how, when, and under what circumstances any particular rule, formula, or process should be applied; and whenever possible one or more examples, such as would be likely to arise in actual practice—together with their solutions—are given to illustrate and explain its application.

In preparing these textbooks, it has been our constant endeavor to view the matter from the student's standpoint, and to try and anticipate everything that would cause him trouble. The utmost pains have been taken to avoid and correct any and all ambiguous expressions—both those due to faulty rhetoric and those due to insufficiency of statement or explanation. As the best way to make a statement, explanation, or description clear is to give a picture or a diagram in connection with it, illustrations have been used almost without limit. The illustrations have in all cases been adapted to the requirements of the text, and projections and sections or outline, partially shaded, or full-shaded perspectives have been used, according to which will best produce the desired results. Half-tones have been used rather sparingly, except in those cases where the general effect is desired rather than the actual details.

It is obvious that books prepared along the lines mentioned must not only be clear and concise beyond anything heretofore attempted, but they must also possess unequaled value for reference purposes. They not only give the maximum of information in a minimum space, but this information is so ingeniously arranged and correlated, and the

indexes are so full and complete, that it can at once be made available to the reader. The numerous examples and explanatory remarks, together with the absence of long demonstrations and abstruse mathematical calculations, are of great assistance in helping one select the proper formula, method, or process and in teaching him how and when it should be used.

The first part of this volume treats on the construction, care, and management of marine boilers and their accessories, the usual methods of firing, and the principles underlying the economic combustion of coal. Special attention has been given to the subject of marine-boiler inspection under United States laws, the rules of the Canadian Board of Steamboat Inspection, and the rules of the British Imperial Board of Trade, as a good knowledge of this subject is necessary for a candidate for a marine engineer's license. Following the treatment of boilers is a section on the elements of propulsion of vessels, followed by a section on the theory of refrigeration; these two sections will prove of great value to chief engineers and others interested. The aim throughout has been to present matter that will be of special value to the operating marine engineer rather than to the designer of marine machinery, with a view of giving a fundamental education that will enable persons having the legal practical experience to easily pass examinations for a marine engineer's license.

The method of numbering the pages, cuts, articles, etc. is such that each subject or part, when the subject is divided into two or more parts, is complete in itself; hence, in order to make the index intelligible, it was necessary to give each subject or part a number. This number is placed at the top of each page, on the headline, opposite the page number; and to distinguish it from the page number it is preceded by the printer's section mark (§). Consequently, a reference such as § 16, page 26, will be readily found by looking along the inside edges of the headlines until § 16 is found, and then through § 16 until page 26 is found.

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TYPES OF MARINE BOILERS

INTRODUCTION

DEFINITIONS

GENERAL NAUTICAL TERMS

1. There are numerous terms and phrases used on board ship with which people living on land are not familiar; hence, the meaning of those terms that are apt to be used by the marine engineer will be explained.

Fig. 1 represents a plan view of a vessel. The forward part of the vessel is called the **bow**; the rear part, the **stern.**

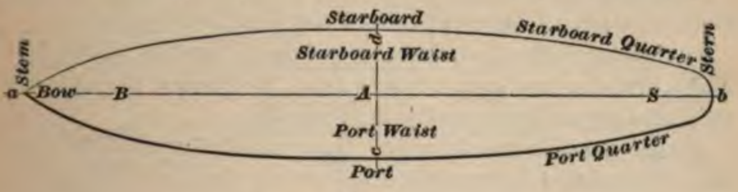

FIG. 1

The forward extremity a of the vessel is known as the **stem.** An observer standing so as to be looking toward the bow has on his right the **starboard** side of the vessel, and on his left the **port** side. Anything located near the center of the vessel, as at A, is said to be **amidship;** any object located near the bow, as at B, **forward;** if located near the stern, as at S, it is said to be **aft.** Any object placed so that

its direction is parallel to the line *a b* is said to be placed **fore and aft**; any object placed so that its direction is at right angles to the line *a b*, as the line *c d*, is said to have an **athwartship** direction. The width of a vessel is called its **beam**. The perpendicular distance from the lowest point of the vessel below the water-line to the surface of the water is known as the **draft**; it is expressed in feet and inches in English-speaking countries. The platforms dividing a vessel into horizontal spaces, forming the ceiling of one space and the floor of the next space above it, are called **decks**. Those parts of the sides of a vessel that project above, and surround the upper deck are called the **bulwarks**, or **rails**. Looking from either rail toward the center line *a b* of the vessel is called looking **inboard**; looking from the center line *a b* of the vessel toward either rail is called looking **outboard**. Any object outside of a vessel that is in line with the athwartship line *c d* is said to be **abeam** or **abreast** of the vessel. When two objects on board of a vessel are in line with each other fore and aft, the one nearer the stern is said to be **abaft** the other one; for example, the engines are abaft the boilers. An object behind the ship is said to be **astern**, and one in front of the vessel is said to be **ahead**.

Beneath the deck is **below**. Ascending from below is *going on deck*. Descending from the deck is *going below*.

Pitching is the alternate up-and-down motion of the bow and stern of a vessel in a rough sea. **Rolling** is the athwartship motion of the vessel in a rough sea.

Windward is the direction from which the wind is blowing. **Leeward** (pronounced lee-ard) is the direction toward which the wind is blowing. When the wind is blowing toward a shore, the latter is known as a *lee shore;* when the wind is blowing in an opposite direction, it is said to be blowing *off shore*. *Under the lee* means being on the leeward side of an elevated object—high land, for instance.

Way is the motion of the vessel through the water.

Leeway is the drift or sidewise motion of the vessel to leeward, driven in that direction by the wind. **Sternway**

is the motion of a vessel when the engines are backing, that is, the going backwards of a vessel. **Under way** is the forward motion of a vessel when running on its course. *Getting under way* is the operation of hoisting anchor or casting off the lines from the wharf and starting the engines.

The **hold** is the cargo or stowage space below deck.

A **hatchway** is an opening in the deck to receive and discharge cargo or stores to and from the hold.

Coal bunkers are the spaces devoted to the stowing of coal. **Trimming the bunkers** is the operation of stowing the coal. **Bunker scuttles** are circular openings in the deck through which the coal is put into the bunkers. **Coaling ship** is taking coal on board and stowing it in the bunkers.

An **ash chute** is an inclined trough through the bulwarks through which the ashes are dumped overboard.

A **hatch** is the cover that is placed over a hatchway when the vessel is at sea. A **companionway** is a hatchway for the ship's company and passengers to descend from or ascend to the deck by means of ladders or stairs.

Shipping a sea is the breaking of a wave over the bulwarks, thus flooding the deck.

A **hatch combing** is a bulwark around a hatchway to prevent the water from going below when a sea is shipped, or while washing the deck.

Waterways are small channels or gutters, around the deck at the base of the bulwarks, to carry off the water.

Scuppers are small openings through the base of the bulwarks to permit the water to flow overboard from the waterways.

Overboard is outside of the vessel, in the sea.

The spaces under the engines, boilers, storerooms, floor plates, etc. are the **bilges**; *pumping bilges* is the operation of pumping out the water collected in the bilges.

Coming to is the act of bringing the vessel to anchor or alongside the wharf. **Laying to** is stopping a vessel while she is on her course to speak another vessel, pick up a pilot, etc. When speaking of a vessel, it is always considered as

having the feminine gender; thus, she is under way, she is laying to, etc.

When a vessel is caught in a violent gale and rough sea, it is sometimes necessary to slow the engines down to **steerage way,** which means to just speed enough to cause the rudder to act on the water sufficiently to control the movements of the vessel and head the vessel into the gale; this is called **heaving to.**

Should a vessel arrive off her port of destination late in the evening, after dark, or during the night and no pilot is obtainable, it is customary to run the vessel slowly back and forth to and from the entrance to the harbor until daylight; this is called **laying off and on. Making fast** is the operation of securing a vessel to a wharf or buoy with hawsers or cables.

Operating the engines in answer to signals is called **working to bells.**

When a screw vessel is pitching violently, the stern rises and lifts the screw propeller out of the water more or less, which causes a sudden increase in the speed of the engine; this is called **racing.**

When a vessel lying in the stream is made fast at both ends, that is, bow and stern, to two buoys or anchors, one ahead and the other astern, she is said to be **moored.**

When a vessel is riding to a single anchor and the tide turns she revolves around a semicircle the center of which is the anchor. This is called **swinging to the tide,** or just **swinging.**

The engineer on duty in the engine room is the **engineer of the watch.**

That part of a vessel or that part of an object on board of a vessel nearest the bow, is called the **forward** part; and that part nearest the stern is called the **after** part, for example, the forward part of the fireroom; the after part of the fireroom; the forward boiler; the after boiler.

The **load water-line** is an imaginary line around the outside of the hull of a vessel that coincides with the water-line when she is fully loaded with cargo, coal, stores, etc. When

the load water-line, forward, is below the surface of the water, the vessel is said to be **down by the head.** When the load water-line, aft, is below the surface of the water, the vessel is said to be **down by the stern.**

When a vessel is immersed in the water more on one side than on the other, she is said to be **listed to port** or **listed to starboard,** as the case may be. When a vessel is immersed in the water equally on both sides, she is said to be **on an even beam.** When a vessel is listed, the act of stowing the cargo or using the coal from the bunkers so that she will be brought to an even beam is called **trimming the ship.**

A **tarpaulin** is a piece of heavy canvas 8 or 10 feet square coated with tar or painted to make it waterproof. In rough weather the hatchways are covered with tarpaulins secured to the hatch combings; this is called **battening down hatches.**

Bulkheads are the partitions in the vessel dividing it into compartments. *Water-tight bulkheads* are tight bulkheads placed in the hull of a vessel, dividing it into water-tight compartments, no two of which are large enough to hold sufficient water, in case of a serious leak, to sink the ship.

When a vessel has a hole knocked in her bottom, she is said to be **stove.**

Ventilators are large sheet-metal pipes, with trumpet mouths placed at right angles with the upright part. They lead from above the deck to the fireroom, hold, etc. to supply these spaces with air. **Wind sails** are large canvas pipes or tubes, with outstretched wings at their tops, leading from above the deck down through the hatchways into the hold to ventilate the ship below decks. When the trumpet mouths of the ventilators or the opening of the wind sails point to windward they are said to be *trimmed to the wind.*

When a vessel is in port and there is no steam on the boilers, the smokestack is often covered with a sheet-metal or canvas covering to keep out the rain; this covering is called the **smokestack hood.**

A **clinometer** is an instrument, usually a pendulum, suspended on a bulkhead, hatch combing, or other convenient place, to designate the number of degrees the ship rolls.

To enlist in the merchant service or in the navy is **to ship**.

An object floating helplessly in the water is said to be **adrift**.

The speed of a vessel is usually measured by means of an instrument called a **log**. This consists of a triangular piece of wood weighted at one side to keep it upright in the water, and called the **log chip**. To the log chip the **log line** is attached, which is coiled on a reel and has pieces of cord tied to it at equal distances apart. To measure the speed of a ship, the log chip is thrown overboard and the log line is allowed to *pay out*, that is, unreel, until the first knot reaches the observer's hand. He then calls out for a second observer to turn over a sand glass, timed to run either 28 or 30 seconds, and at the moment the glass is turned over, lets the log line pay out again, noting the number of knots that pass through the hand while the glass is running. Then the number of knots and fraction thereof counted represent the rate of advance of the vessel either in nautical or in statute miles per hour. Thus, if 12½ knots slip through the observer's hand, the speed of the vessel is said to be 12½ knots, which corresponds to 12½ nautical or statute miles per hour, depending on what mile the log line is divided for.

The nautical mile is in practice taken as 6080 feet, which value has been assigned to it by the British Admiralty, and which has been universally adopted. The statute mile is 5280 feet in length.

For a 28-second glass the knots in the log line are 47.29 feet apart for the nautical mile and 41.06 feet for the statute mile; for a 30-second glass the knots are 50.6 feet apart for the nautical mile and 44 feet for the statute mile.

The nautical mile is used as the unit of distance in ocean navigation, and the statute mile in river, lake, and inland navigation in general.

Landsmen often erroneously speak of the speed of a vessel as being so many *knots per hour;* as has been explained, the term knot defines the rate of speed, but not the distance traversed in one hour. This should be expressed distinctly in nautical or statute miles.

The **displacement** of a vessel is equal to the weight of the water it displaces, and is usually expressed in tons of 2,240 pounds. It will vary with the draft, for the deeper the vessel is in the water, the more water will it displace. The **tonnage** of a vessel is its entire internal cubic capacity, measured in the United States in tons of 100 cubic feet each, in a manner prescribed by law. Tonnage should not be confounded with displacement.

SPECIAL NAVAL TERMS

2. There are certain phrases used in the navy that are not commonly used in the merchant service. The captain's quarters is called the **cabin.** The cabin on board a naval vessel is located aft. The commissioned officers' quarters is called the **ward room,** which is usually located just forward of, or underneath the cabin, according to the construction of the vessel. Just forward of the ward room is the **steerage,** where the warrant and appointed officers and midshipmen are quartered. The crew is quartered on the **berth deck,** which is located forward of the engines and boilers. The hospital of the ship is called the **sick bay,** and is usually located forward of the berth deck at the bow. The ship's prison is called **the brig.** The **quarter deck** is the starboard side of the main deck abaft the mainmast when the vessel is in port, and the windward side at sea. It is only occupied by the captain, the executive officer, and the officer of the deck. Everybody else is supposed to keep off unless they have business with one of the officers mentioned, and after the business is transacted they are expected to depart immediately for their own part of the deck. All persons on entering on the quarter deck are required to touch their caps with the fingers of the right hand; this is called *saluting the quarter deck.*

At the bow, there is often a small deck elevated above the main deck, and called the **topgallant forecastle** (pronounced t-gallant-fo-cassel); on this, in fine weather, the crew congregate during recreation hours to smoke. During foul weather, they smoke under the topgallant forecastle deck. The **waists** of the ship are the passageways between the bulwarks and the hatch combings, on each side of the deck. That on the port side is called the *port waist;* that on the starboard side is called the *starboard waist.* The **gangways** are openings cut in the bulwarks of the vessel for the entrance to and exit of persons from the deck, to or from boats or the wharf.

The **warrant machinists** in the United States navy now act as assistant engineers. They operate the engines and have charge of a watch. There is another position in the engineer's force of a man-of-war, called the **engineer's yeoman.** He has charge of the tools and stores (supplies), and serves them out when they are needed. He also copies the log, writes the reports, and keeps the expenditure book, and acts, in general, as the chief engineer's clerk. This position seldom, if ever, exists in the merchant service, the nearest approach to it being that of **storekeeper,** or man in charge of the stores and tools.

GENERAL DESCRIPTION OF A BOILER

ESSENTIAL PARTS

3. A **steam boiler** is an apparatus for the generation of steam from water for various industrial purposes, such as the production of mechanical power to operate machinery or propel vessels through the agency of the steam engine, and for heating and drying purposes. A boiler must contain three essential parts, which are: (1) a place for the fire, (2) a place for the water, and (3) a division or partition between them.

A steam boiler consists of a vessel containing water, which is converted into steam by the application of heat. The heat is generated by the combustion of some fuel, such as coal,

wood, petroleum, etc., in the **furnace.** To carry away the products of combustion and to create a **draft,** that is, to supply the burning fuel with air, the furnace is connected with the **smokestack,** sometimes called the *funnel.* The water used for the generation of steam is supplied to the boiler by the **feed-apparatus,** and enters the boiler through the **feedpipe.** The steam generated in the boiler is conveyed to its destination by the **steam pipe.**

When any portion of a boiler, such as a plate or tube, is in contact with the fire or hot gases on one side and water on the other, the surface in contact with the fire and hot gases is called a **heating surface.** The sum of all such surfaces is called the *total heating surfaces.* Such surface as is in contact with fire or hot gases of combustion on one side and steam on the other side is called a *superheating surface.*

The furnace is provided with a **grate,** on which the fuel is placed to be burned. The grate usually consists of a series of cast-iron bars with spaces between them for the admission of air to the burning fuel. The area of the grate, expressed in square feet, is called the *grate surface.*

It is imperative that a steam boiler should only be partly filled with water when ready for service. As 1 cubic inch of water occupies nearly 1 cubic foot of space when converted into steam at the atmospheric pressure (but less at higher pressures), a very considerable portion of the space within a boiler must be reserved as a receptacle or reservoir for the steam; this is called the **steam space,** and, as a matter of course, it is located at the highest part of the boiler above the water-line. That portion of a boiler occupied by the water is called the **water space.**

CLASSIFICATION

4. Steam boilers may be divided into four distinct classes or types, namely: stationary, portable, locomotive, and marine boilers. The first three classes may be grouped under the general head of *land boilers* to distinguish them from those used on vessels, which are termed *marine boilers.*

Marine boilers are divided into two distinct types: *fire-tube boilers*, and *water-tube*, or *tubulous*, *boilers*. Their distinguishing features are: In **fire-tube boilers**, the flame and gases of combustion pass through tubes or flues which are surrounded by water; whereas, in **water-tube boilers**, the water circulates through the tubes and the flame and gases of combustion surround them. Another distinguishing feature is that in fire-tube boilers the tubes and flues are enclosed in a shell, which must be strong enough to sustain the steam pressure within it, while the tubes of the water-tube boilers are enclosed in a casing of light sheet-iron lined with some refractory and non-heat-conducting substance, such as asbestos, magnesia, mineral wool, etc. This casing is not called on to sustain any pressure, that duty being performed by the tubes, steam drums, and mud-drums, which, being of small diameter compared with the shell of tubular or flue boilers, may be made of much thinner plates, and consequently lighter, than the shell of fire-tube boilers.

CONSTRUCTION OF MARINE BOILERS

FIRE-TUBE BOILERS

FLUE BOILERS

5. Externally Fired Flue Boilers.—The simplest form of marine boiler, as used at the present time, is the **flue boiler**, shown in Fig. 2. This type of boiler is still in extensive use on Western-river steamboats. It consists essentially of a long cylinder *a*, called the **shell**, made of iron or steel plates riveted together. The ends of the boiler are closed by flat or hemispherical plates *b*, *b*, called the **heads** of the boiler. Two or more **flues** *c*, *c*, in some instances as many as six, are fixed to the front and rear heads. To the front of the boiler a sheet-iron casing *d*, known as the **front connection**, is secured; the upper part of this leads to the **smokestack** *e*. To give access to the

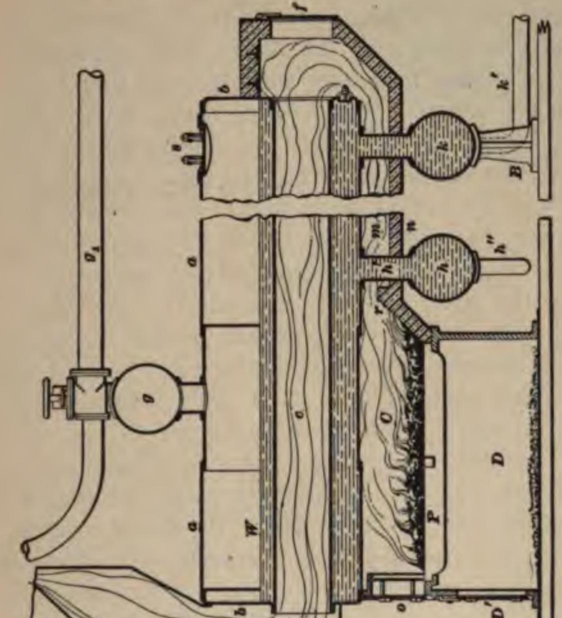

FIG. 2

flues, doors f are provided. When two boilers are placed in one setting, as shown in the figure, they are usually connected to a **steam drum** g, the object of this drum being to furnish dry steam. Fitted to the steam drum is the stop-valve g'', by means of which communication between the boilers and the engines may be shut off. Connected to the stop-valve is the steam pipe g_1 that conveys the steam to the engines. Attached to the top of the steam drum is the **safety valve** g', which prevents the steam pressure from exceeding the safe working pressure of the boiler. To indicate the steam pressure, a **steam gauge** t is attached to each set of boilers. To indicate the water level within the boiler, **gauge-cocks** l are fitted to the front heads. A **manhole** s, which is simply a hole cut in the shell and closed by a suitable cover, gives access to the inside of the boiler. To provide a quiet place for the settlement of the foreign matter held in suspension in the water used for feeding the boiler, a **mud-drum** h, connected to the shell by the nozzle h', is provided. Attached to the mud-drum is a blow-off pipe h'' provided with a stop-cock, not shown in the figure, by means of which the sediment may be drawn off, or the boiler emptied. A similar drum k is attached to the rear of the boilers to receive the feedwater, which passes thence into the boilers. The pipe k' leads to the feed-pump.

As usually set, the front ends of the boilers are supported in a cast-iron front A resting on the deck. The rear ends are supported by cast-iron brackets B placed underneath the feed-drum and secured to the deck. Brickwork, lined with firebrick, forms the sides and top of the boiler setting. The bottom of the setting that forms the lower smoke flue m is made of wrought-iron plates n, n lined with firebrick. The furnace C is placed under the front end of the boiler shell. The fuel is thrown in through the **furnace doors** o, o and burns on the **grate** P, the ashes falling through the grate into the **ash-pit** D, which is provided with doors D'. Behind the furnace is built the firebrick **bridge** r. It serves to keep the hot gases in close contact with the under side of the shell. The gases arising from the

fire doors *f* are provided. When two boilers are placed in one setting as shown in the figure, they are usually connected to a **steam drum** *g*, the object of this drum being to furnish dry steam. Fitted to the steam drum is the stop-valve *g'*, by means of which communication between the boilers and the engines may be shut off. Connected to the stop-valve is the steam pipe *g*, that conveys the steam to the engines. Attached to the top of the steam drum is the **safety valve** *g*, which prevents the steam pressure from exceeding the safe working pressure of the boiler. To indicate the steam pressure, a **steam gauge** *i* is attached to each set of boilers. To indicate the water level within the boiler, **gauge-cocks** *l* are fitted to the front heads. A **manhole** *s*, which is simply a hole cut in the shell and closed by a suitable cover, gives access to the inside of the boiler. To provide a quiet place for the settlement of the foreign matter held in suspension in the water used for feeding the boiler, a **mud-drum** *k*, connected to the shell by the nozzle *k'*, is provided. Attached to the mud-drum is a *blow-off pipe* *k''* provided with a stop-cock, not shown in the figure, by means of which the sediment may be drawn off, or the boiler emptied. A similar drum *k* is attached to the rear of the boilers to receive the feedwater, which passes thence into the boilers. The pipe *k'* leads to the feed-pump.

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combustion of the fuel pass from the furnace *C* over the bridge *r* into the smoke flue *m*, thence into the cylindrical boiler flues *c*, *c*, whence they pass into the front connection *d* and up the smokestack, which is provided with a **damper**, shown at *e′*.

It will be seen, by referring to Fig. 2, that the brickwork of the setting covers the upper portion of the boiler shell in such a manner as to prevent the hot gases from coming in contact with the shell above the water-line *W*. The part of the boiler shell above the brickwork, the steam drum, and the steam pipe are covered with some non-conducting material to prevent the radiation of heat.

In the flue boiler, the heating surface, according to the definition, consists of the part of the shell that is over the furnace, the rear head (both to be measured up to the under side of the top of the setting), the inside of the flues, and the front head. The front head is usually omitted in calculating the heating surface.

6. A typical modern Western-river steamboat boiler plant is shown in Fig. 3. Fig. 3 (*a*) is a front elevation, looking aft. Fig. 3 (*b*) is a side elevation, looking from port to starboard. Fig. 3 (*c*) is a plan view, a section being taken through the port boiler along its horizontal center line in order to show the flues and the inside of the setting. Fig. 3 (*d*) is a rear view of the boilers and setting, part of the setting being broken away around the starboard boiler, through which a vertical section has been taken in order to show the location of the flues. The three boilers *A*, *B*, and *C* are of the flue type, containing five flues. The location of the flues is shown in Fig. 3 (*c*) and (*d*). There is one wide grate common to all boilers; when there are more than three boilers in a battery, there may be two or more furnaces. The breeching, or front connection, *D*, *D* is common to all boilers, and is provided with doors, as shown, to allow the flues, etc. to be examined and cleaned. The boilers are externally fired, and the gases of combustion surround about two-thirds of the shell. They pass to the rear of the boiler,

and then through the flues forwards again into the breeching, whence they pass up the two stacks E, E. The use of two stacks is common on all Western-river steamers, as it gives the pilot an unobstructed view forward and aft. When only one stack is used, as is sometimes done in the smaller vessels, it is usually set on one side of the boat, so as not to obstruct the view of the pilot. Each boiler is provided with its own safety valve; the nozzles to which the valves are attached are shown at a, b, and c. Occasionally two safety valves are used, one of them being a lock-up safety valve, set by the boiler inspector to the steam pressure allowed, and the other a common safety valve. The three boilers are connected by suitable flanged nozzles to the steam drum F, forming a steam reservoir. The main steam pipe leading to the engines is connected to the nozzle f. All other steam pipes, such as those for the whistle, steering gear, feed-apparatus, capstan, etc., are also connected to the steam drum at suitable places. The bottoms of the three boilers are connected together by two mud-drums, or **stand pipes**, as they are often called; the rear mud-drum is shown at G and the forward mud-drum at H. These mud-drums are supposed to provide a quiet place for the collection of mud and sediment held in mechanical suspension in the feedwater. Each drum is provided with two nozzles to which the **mud-valves** or blow-off valves are attached. Suitable pipes lead the water overboard. The nozzles g, g of the rear mud-drum are attached to the lower part of the drum and point aft. The nozzles on the forward drum are attached to the lower part of the two heads. One of these nozzles is shown at h. The nozzle j in the center of the rear mud-drum is for the donkey feed-pipe. Each boiler has its own main feedpipe and check-valve; the water is introduced through the rear head, and passing through a coil of pipe is delivered in the steam space near the front of the boilers. The gauge cocks i are in the rear head of the boiler, as are also the float water gauges o. The purpose of placing the gauges in the rear head of the boilers is to allow the engineer to see the height of water in the boilers without leaving the engine room.

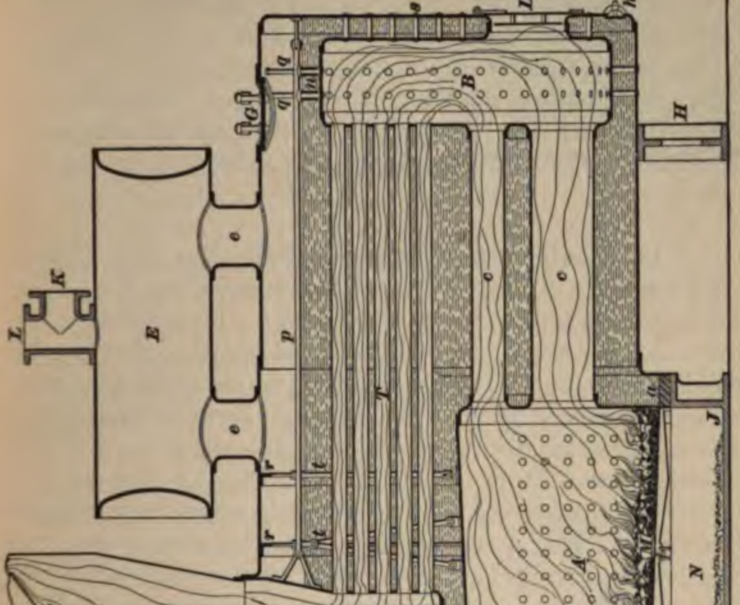


FIG. 4

Suitable manholes and handholes are provided to allow examination and repair of the boilers, mud-drums, and steam drums. The front of the boiler setting is of cast iron; the sides, rear, bottom, and top of sheet iron. Every part of the setting that is exposed to the fire is lined with firebrick. The boilers and setting are secured to the deck by the tie rods *n*, *n*.

7. Internally Fired Flue Boiler.—The flue boilers shown in Figs. 2 and 3 have the furnace outside of the boiler and hence are called **externally fired boilers.** The desire for more compact and self-contained boilers, that is, for boilers requiring no brick setting, led to the development of **internally fired boilers,** in which the furnace is contained within the boiler itself. A boiler of this class, known as a **firebox flue boiler,** is illustrated in Fig. 4. The shell of the boiler is composed of two differently shaped parts riveted together. The rear part of the boiler is cylindrical; the front part is of a rectangular cross-section with vertical sides and a semicircular top. There are one or more furnaces *A* (two in this case), with vertical sides and a round top. A space is left between the two furnaces as well as between the furnaces and the sides of the boiler; these spaces, shown at *a*, *a*, are filled with water, and are known as the **water legs.** From the furnaces, the large flues *c*, *c*, *c* lead to the combustion chamber, or **back connection,** *B* common to both furnaces. Two nests of tubes *T* connect the combustion chamber with the front connection, or uptake, *C*. It will be noticed that the uptake is inside the boiler and surrounded by water at the lower end and by steam at the upper end. Such an uptake is called a **wet uptake** to distinguish it from the form of an uptake placed entirely outside of the boiler, and known as a **dry uptake.** Access to the front and back connections is provided by means of the doors *D*, *D'*. The upper part of the uptake opens directly into the smokestack *S*, which is provided with a damper *F*. The steam drum *E* is connected to the shell by the nozzles *e*, *e*, as shown. The manhole is shown at *G*. Handholes *h*, *h*, *h* are provided to

facilitate the cleaning out of the water legs. The boiler is supported at the rear by the cast-iron saddle *H*, to which it is firmly bolted, the saddle in turn being securely attached to the timbers or framing of the bottom of the vessel. The front part of the boiler is fastened to a cast-iron frame, shown at *J*. This frame is bolted to the framing, and also forms the ash-pit *N*. Since the flat sides of the furnaces and shells would bulge on account of the pressure, they must be braced or stayed; this is accomplished by the screw stays *s*, *s*. Similar screw stays are employed to connect the combustion chamber with the rear head, strengthening them both against bulging; this kind of stay is also used between the combustion chamber and the outer shell except at the top, which is strengthened by the girder stays *n* supported by the sling stays *q*, *q*. The top of the furnace is stayed by the toggle braces *t*, *t*, attached to rings, shown at *r*, *r*, made of angle iron and riveted to the shell. Toggle braces *p* are used to stay the flat surfaces of the rear head and of the uptake. At *K*, the steam pipe is attached; at *L*, the safety valve; the furnace door is shown at *P*; the grate at *M*. The gases of combustion pass from the furnaces *A* through the flues *c* to the combustion chamber *B*, whence they pass through the tubes *T* to the front connection *C* and up the smokestack *S*. As shown in this figure, the water legs only extend a little below the grate, the ash-pit being formed by the frame *J*. Sometimes there is a water space below the ash-pit; that is, the furnace and ash-pit are entirely surrounded by water; a boiler constructed in this manner is known as a **wet-bottomed boiler.** The type illustrated in Fig. 4 is called a **dry-bottomed boiler.**

In the firebox boiler shown, the heating surface is formed by the inside of the furnaces above the grate, the inside of the flues *c*, the sides, top, and bottom of the combustion chamber, deducting, of course, the spaces taken up by the flues *c*, *c* and the door *D'*, the inside of the tubes, and the sides and bottom of the uptake. Only in case of a wet uptake is its surface to be taken as heating surface, and then only up to the water-line.

FIREBOX TUBULAR BOILERS

8. Wet-Bottomed Firebox Tubular Boiler.—By extending the principle of the flue boiler, that is, reducing the size and increasing the number of flues, the tubular boiler is evolved. This boiler, which gives a greater heating surface than the flue boiler, is constructed in various forms, varying chiefly in small details. A pair of **wet-bottomed firebox tubular boilers** is shown in Fig. 5. Boilers of this kind are used to some extent on steamboats navigating the Western rivers of the United States of America, and in similar service. Fig. 5 (*a*) is a front view looking forwards, and Fig. 5 (*b*) a vertical fore-and-aft section through the starboard boiler. Each boiler is composed of two parts, the forward part being cylindrical and the after part approximately rectangular with a semicircular top. There are two furnaces in each boiler, each furnace having its own nest of tubes leading to the front connection *A*, which is common to both furnaces. The steam generated in the boilers passes through the nozzles *a* and *b* into the steam drum *B*. The main steam pipe is connected at *c* to the steam drum. The auxiliary steam pipes are also connected to this drum. The water space of both boilers is connected by the mud-drum *C*. In this design, each boiler has its own front connection and smokestack, although more than one boiler may be connected to one stack. Each boiler has its own safety valve, which is attached at *d*. A manhole is shown at *f* and some handholes at *i*; these are placed in various parts of the boiler to allow it to be examined and cleaned. Suitable manholes and handholes are also provided for the steam drum and mud-drum. As shown in the figure, the furnaces are surrounded entirely by water, hence this boiler belongs to the wet-bottomed type. The flat surfaces of the boiler are stayed by the screw stays *l* and the diagonal braces *m*. The crown sheets of the furnaces are stayed by the crown bars *o* made of **T** iron, which are riveted to the crown sheets by numerous rivets. Distance pieces or thimbles are placed between the top of the crown sheet and

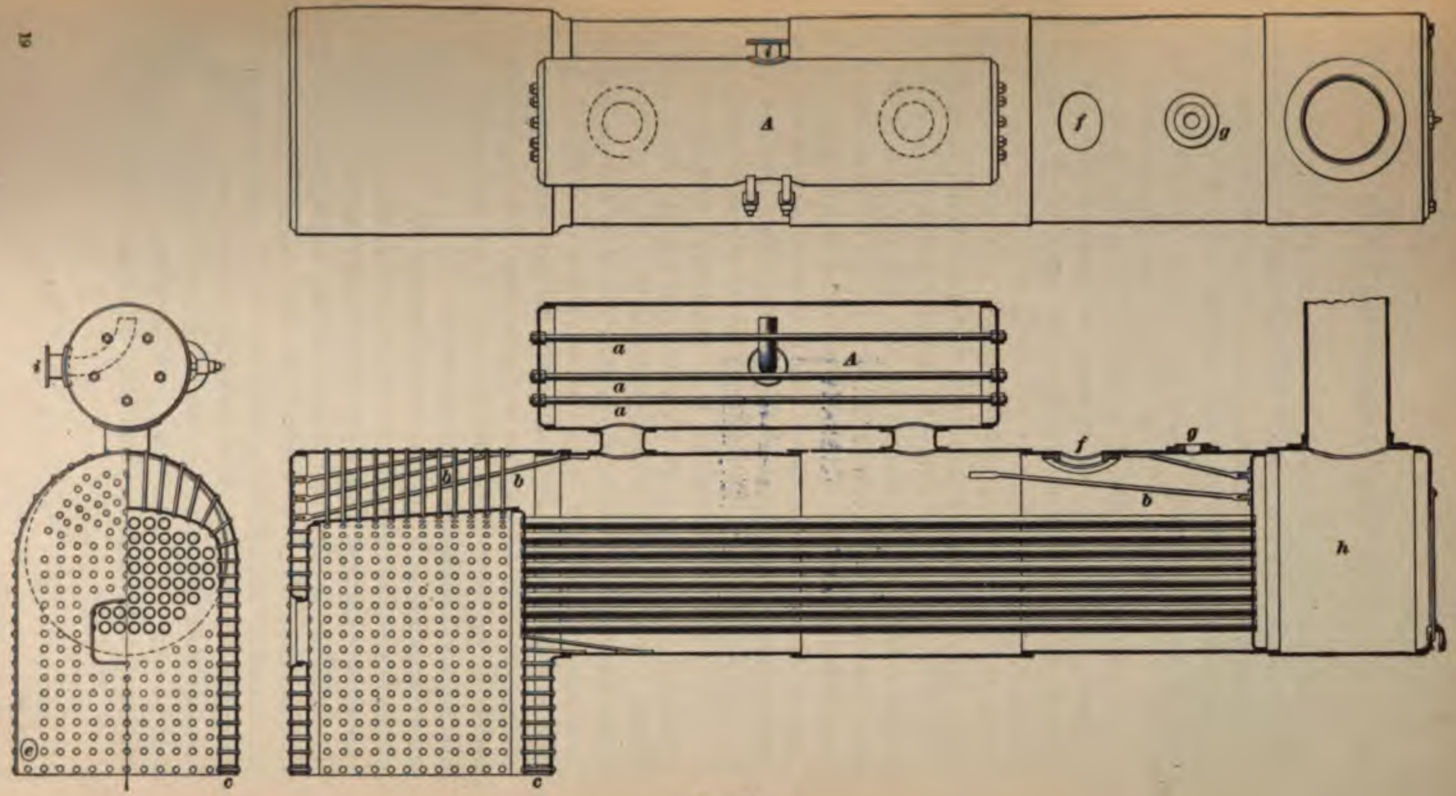


FIG. 6

the bottom of the crown bars. The crown bars are connected to the top of the boiler by the toggle braces *n*, which are cottered to the crown bars and riveted to the shell. In the particular design of boiler shown, a bridge wall *E* is built in each furnace. However, firebox boilers are frequently built without a bridge wall. The smokestacks are supported by the stanchions shown. The boilers are secured to the deck by tie-rods, not shown in the illustration. This style of boiler being internally fired, there is no elaborate setting required for them.

9. Dry-Bottomed Firebox Tubular Boiler.—Fig. 6 illustrates a form of a firebox tubular boiler that is constructed with a dry bottom to the ash-pit, and from its resemblance to the boiler of a locomotive is often called a **locomotive boiler.** This type of boiler is principally used in river and similar service. The boiler consists of two differently shaped parts riveted together. The forward part is cylindrical; the cross-section of the after part is rectangular with a semicircular top. It has one large furnace, which is surrounded by water on the sides and top, but open at the bottom, thus making it a dry-bottomed boiler. It is provided with a steam drum *A*, the heads of which in the larger sizes are stayed by through stayrods *a*, *a* provided with nuts on the inside and outside of the heads. The ash-pit is entirely separate from the boiler proper, the grate being placed at the bottom of the furnace. The flat surfaces of the furnace and also the crown sheet are stayed by screw stays screwed into the sheets and riveted over. That part of the rear head which is not stayed to the furnace plate is stayed by diagonal stays *b*, *b*, similar stays being employed for the front head. The bottom of the water legs, as the spaces surrounding the furnace are often called, is closed by a cast-iron or wrought-iron mud-ring *c*. The water legs are provided with handholes in suitable locations; one of these handholes is shown at *e*. At *f* a manhole is shown. The safety valve is attached at *g*. The gases of combustion traverse a nest of tubes extending from the rear tube sheet

to the front head, and are discharged into the front connection h, whence they pass up the stack. The main steam pipe is connected to the nozzle i on the side of the steam drum; a curved pipe takes the steam from the top of the drum.

SCOTCH BOILERS

10. Single-Ended Scotch Boiler.—For seagoing steam vessels a type of internally fired tubular boiler has been gradually developed that is known commonly as the **Scotch boiler,** and is occasionally spoken of as the **drum boiler.** This type of boiler is used in the service mentioned practically to the exclusion of all other types of fire-tube boilers, and it is only within recent years that boilers of the water-tube type have to a considerable extent taken its place.

A **single-ended Scotch boiler** is shown in Fig. 7. By single-ended is meant that the boiler has furnaces at one end only. It has a cylindrical shell with flat heads. The diameter of these boilers may vary from 10 to 15 or even 20 feet; the length may vary from 7 to 11 feet. The boiler is provided with two, three, or four large corrugated furnace flues A, A. The one shown in the figure has four. The rear end of each furnace flue opens into a **combustion chamber** B. Usually each flue has its own combustion chamber, but in some cases two or more flues open into a common chamber. Nests of tubes T, T extend from the front plate of the combustion chamber to the front head of the shell. These tubes place the combustion chambers B, B in communication with the large smoke chamber E, commonly known as the front connection. The upper part of this chamber forms the uptake, which in turn leads directly to the smokestack. The flat heads of the shell are kept from bulging by the heavy stayrods R, R, and further by the diagonal braces, or **palm stays,** H, H. About one-third of the tubes (those marked with a cross in the figure) are threaded and provided with nuts, and thus act as stayrods for the flat surfaces occupied by the tubes. The flat sides of the combustion chambers are stayed to each other and to the rear head by

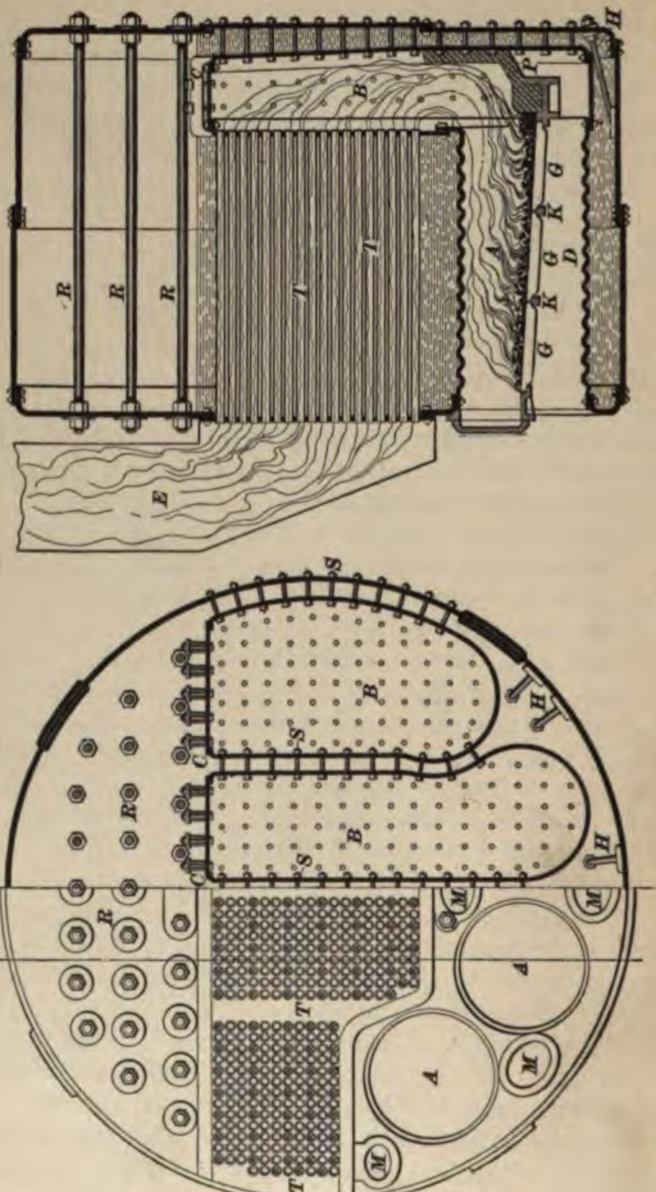


FIG. 7

the staybolts *S*, *S*. The flat tops of the combustion chambers, called the **crown sheets**, are strengthened by the **girder stays**, or **dogs**, *C*, *C*. The manholes *M*, *M* give access to the various parts of the boiler. The various fittings are not shown in the figure, but are attached in convenient places. The furnaces are placed within the corrugated flues *A*, *A*. As shown, the grate *G* is made in three sections, supported by the cross-bars *K*, *K*. Below the grate is the ash-pit *D*. At the rear end of the grate is placed a firebrick plate *P* for the purpose of preventing cold air from sweeping through the ash-pit into the combustion chamber without first passing through the grate. The gases arising from the combustion of the coal pass into the combustion chamber *B*, where they undergo further combustion in contact with the air that passes through the grate. The hot products of combustion then pass through the tubes to the front connection *E*, and out through the smokestack. The flues and tubes are completely surrounded by water; likewise, the combustion chambers.

11. In Art. **3**, it was stated that the heating surface of a boiler is the surface that is in contact with the fire or hot gases of combustion, the other surface of the plate, etc. being in contact with the water. In Fig. 7, the water is shown covering the tubes *T*, *T* to a depth of several inches. Hence, from the foregoing definition, the heating surface of a Scotch boiler consists of the following: (1) The part of the furnace flues above the grate; (2) the back, sides, and top of the combustion chamber; (3) the front plate of the combustion chamber, known as the **back tube-sheet**; (4) the inner surface of the tubes; (5) the part of the front head pierced by the tubes, known as the **front tube-sheet**. As the front tube-sheet does not form a very efficient heating surface, it is customary not to consider it as such in computing the heating surface of a boiler.

12. Double-Ended Scotch Boiler.—In Fig. 8 is shown a section of a **double-ended Scotch boiler**. At the left end of the figure, the section is taken through one of the

lower flues, and at the right end through one of the upper flues. The furnace flues are placed in each end of the boiler, the combustion chambers being near the center. Each chamber is connected with the nearest head by a nest of tubes leading to the uptakes *H*, *H*. The flat surfaces are stayed and braced as just described in the case of the single-ended boiler shown in Fig. 7. The process of combustion and the path of the gases are the same as for the single-ended boiler. The plates *P*, *P* prevent air from passing from the ash-pit to the combustion chamber. The steam dome, or drum, is attached at *D*.

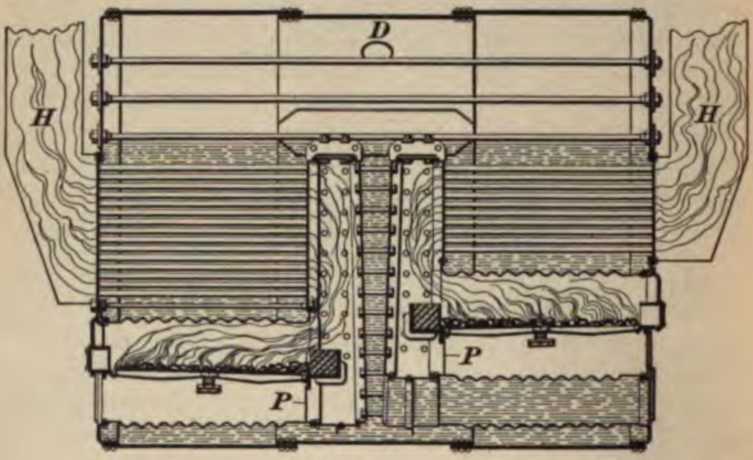


FIG. 8

13. It will be observed that a double-ended boiler is practically two single-ended boilers with their back heads removed and the shells placed back to back and joined. By avoiding the use of the two back heads, a large saving of weight is effected, and the cost of construction is less. Therefore, a double-ended boiler is lighter, cheaper, and occupies somewhat less space in length than two single-ended boilers of the same aggregate steaming capacity.

To still further lighten double-ended boilers, common combustion chambers for corresponding furnaces at the two ends have been used. Such a form of double-ended boiler

is shown in Fig. 9. The furnace flues are placed in each end of the shell, but each opposite pair opens into a common combustion chamber *C*. Each of these combustion chambers has two nests of tubes, one nest connecting it with one head, the other nest with the other head. The gases from two opposite furnaces mix together in the common combustion chamber and then pass through the two nests of tubes, one-half to one smoke flue, the other half to the other. In other respects, the construction of the boiler is similar to that shown in Fig. 8. The steam dome is attached at *S*.

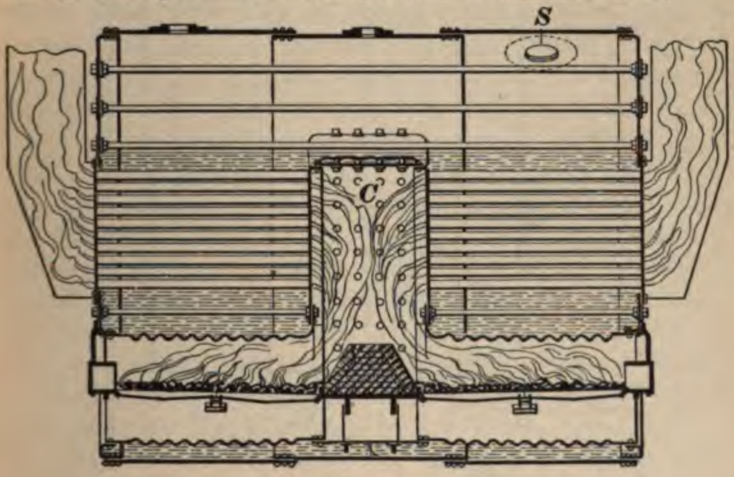


FIG. 9

14. Clyde Boiler.—A modification of the Scotch boiler, and, in outward appearance, resembling it, is shown in Fig. 10. The difference between the two boilers is in the construction of the combustion chambers. In the Scotch boiler, the combustion chamber is surrounded by water; while, in the boiler shown in Fig. 10, the combustion chamber is not surrounded by water and is simply attached to its rear head. For this reason, it is often called a **dry-back Scotch boiler**, although most engineers refer to it as the **Clyde boiler**, presumably because this type was originally designed in the shipyards on the river Clyde, Scotland. The object of this modification is to reduce the weight and cost of

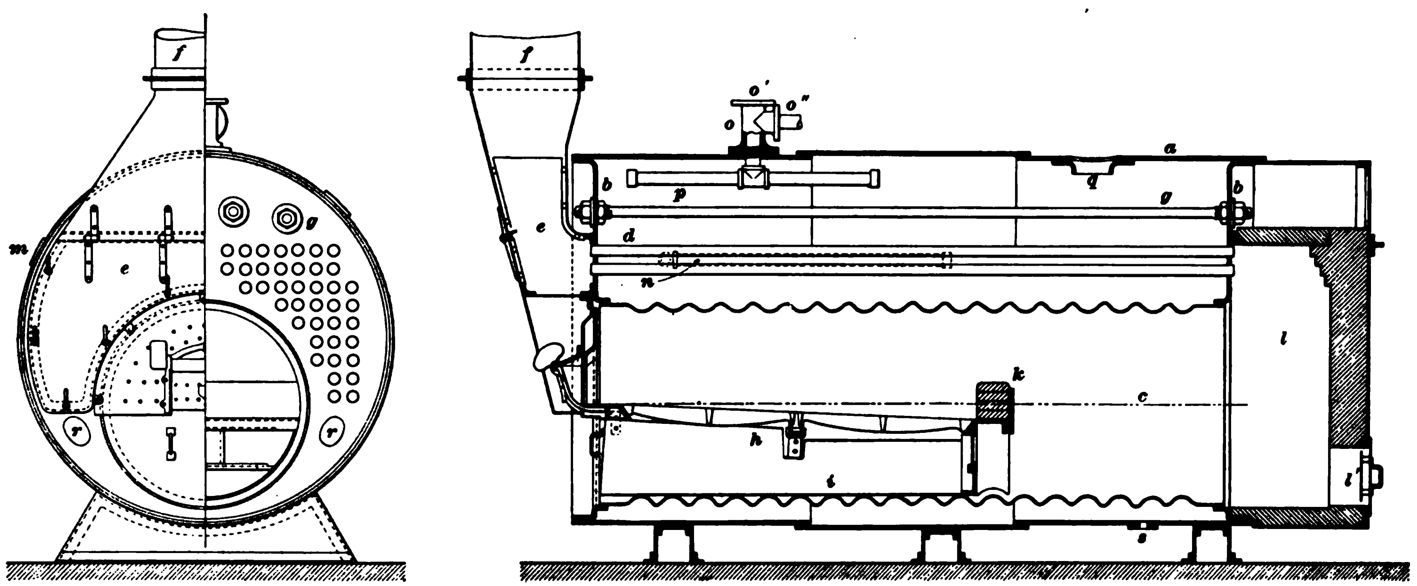


FIG. 10

construction. It provides a light and inexpensive boiler for small and moderate-size vessels, tugs, and like craft.

A boiler of this type consists of a large cylindrical shell *a*, the ends of which are closed by the flat heads *b*, *b*. A large furnace flue *c* of the type known technically as the **Morison suspension furnace flue,** extends clear through the boiler and is securely riveted to the two heads, which are flanged inwards for this purpose. Above and beside the furnace flue, and parallel thereto and below the water-line, is a nest of tubes *d* that extend from head to head. The front ends of these tubes open into an uptake *e* that connects with the chimney or stack *f*. The flat heads are stayed by through stayrods *g*, *g* in the steam space, which prevent deflection of the heads. The remaining parts of the flat heads are supported by the tubes, which are expanded and beaded over, and by the furnace flue. The furnace is placed within the furnace flue, and, as usual, consists of the grate *h*, the ash-pit *i*, and the bridge *k*. The gases of combustion flow to the rear into the combustion chamber *l* and then pass through the tubes to the front and into the uptake. The combustion chamber is formed by a thin cylindrical shell attached to the rear end of the boiler, and is lined with firebrick or thick asbestos millboard, which is light and is not affected by intense heat. The back plate is removable, giving access to the rear ends of the tubes. A door l' gives access to the combustion chamber for the removal of ashes and soot and for the purpose of examination and repair. The feedwater enters the boiler at *m* and, passing through the internal perforated feedpipe *n*, is discharged downwards alongside the shell in small streams. The various fittings are not shown in the illustration. The steam gauge and water column would naturally be located close to the front end of the boiler; the safety valve is intended to be bolted to the outlet o' and the steam pipe to the outlet o'' of the nozzle *o*. The steam is collected by the dry pipe *p*, which is perforated with numerous slots on top. The dry pipe is fairly effective in freeing the steam from any water that may be mixed with it. The manhole is at *q* and two handholes at *r*. The blow-off is

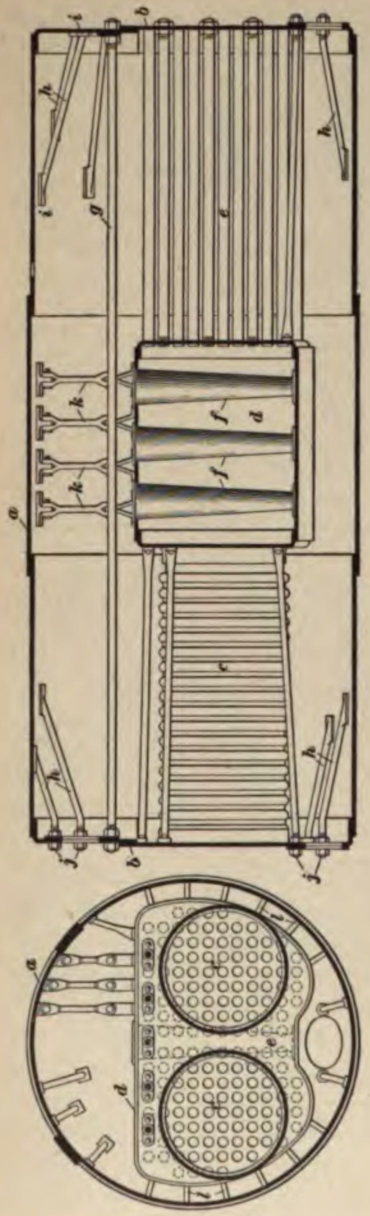


FIG. 11

attached at s. The boiler is entirely self-contained, that is, it does not require any brickwork setting. It is simply bolted to three saddles that rest on and are fastened to the framing of the vessel.

15. Gunboat Boiler. A modification of the Scotch boiler, made for the purpose of providing a boiler of small diameter that can be placed in gunboats and other small light-draught vessels not having space enough below the deck for the regulation Scotch boiler, is the **gunboat boiler,** shown in Fig. 11. The peculiarity of this boiler is that the tubes, instead of being placed above and around the furnace flues, are placed in the rear and in line with them. By this arrangement of the parts, the boiler is greatly reduced in diameter, but its length is doubled. The reduced diameter enables the shell to be made of thinner plates. This boiler consists of the cylindrical shell a with flat heads b, b. The corrugated furnace

flues *c, c* are similar to those used in the ordinary Scotch boiler, and, as usual, contain the grates. The combustion chamber *d* is made twice the depth of the combustion chamber of a Scotch boiler of the same capacity, to compensate for its reduced height. The tubes *e* extend from the rear wall of the combustion chamber to the rear head of the boiler. The uptake or smokebox (not shown in the illustration) leading to the smokestack is attached to the rear head of the boiler. The combustion chamber is provided with the vertical tapering tubes *f, f, f*. These connect the upper and lower parts of the water space together, promote circulation, add considerably to the heating surface, and assist in staying and strengthening the flat top of the combustion chamber. They are made tapering to enable the flange at the lower or smaller end of the tube to be passed through the opening in the top sheet of the combustion chamber while the boiler is under construction. The tapering form, with the large end uppermost, also facilitates the release and discharge of the steam that is generated within the tubes, which are called **Galloway tubes.** The heads are braced by the tubes *e*, the furnace flues *c, c*, the longitudinal braces *g*, and the diagonal braces or palm stays *h, h*. The palm stays are made of round bar iron or steel with flat pieces welded to them. In some cases, they have a palm *i* at each end, which are riveted to the shell and the head of the boiler; in other cases, they have a palm at one end only and are threaded at the other end. When they are made in this way, the palm end is riveted to the shell of the boiler and the threaded end passes through the head, with a nut on each side of the plate, as shown at *j*. The flat top of the combustion chamber is braced by the sling stays *k, k*. The sides and bottom of the combustion chamber are secured to the shell of the boiler by the staybolts *l, l*.

16. Combustion Chambers of Scotch Boilers.—The variations of each type of Scotch boiler are usually in their combustion chambers, the single-ended variations being as follows: (1) each furnace has a separate combustion

chamber; (2) two or more furnaces have a common combustion chamber. Double-ended variations are: (1) each furnace has a separate combustion chamber; (2) two or more furnaces in one end of the boiler have a common combustion chamber; (3) two or more furnaces in line with each other in opposite ends of the boiler have a common combustion chamber.

The relative merits of each of these variations are still in dispute, but practice seems to show that a separate combustion chamber for each furnace is the best arrangement. The separate combustion chamber variation is the one adopted by most of the builders of large boilers in the United States of America. The advantages of having a separate combustion chamber for each furnace are: (1) the circulation of the water is much improved, and the boiler, in consequence, is kept at a more uniform temperature; (2) repairing and overhauling can be performed more easily in the separate nests of tubes; (3) if forced draft is used, it is more efficient and better combustion is insured; (4) working on one fire does not cool down the others or interfere with the draft; (5) the boiler is stronger when made from this design than if the combustion chamber is common to several furnaces.

About the only disadvantages of this method of construction are that the boiler is heavier and costs more to build.

VERTICAL MARINE BOILERS

17. Purpose.—Although **vertical boilers** are becoming obsolete for marine purposes, there are still many of them in use in steam launches, and in larger vessels as auxiliary or donkey boilers. They are not economical in the use of fuel, as, on account of the short passage of the gases of combustion from the furnace to the stack, a considerable amount of the heat of combustion escapes up the smokestack and is lost. These boilers are, however, generally made in small sizes, in which economy of fuel is not of so much importance as in larger boilers. Unless they are built very low, they are top heavy, which raises the center

of gravity of small craft, making them unstable. On the other hand, they should not be too low, else the tubes will be too short and the passage of the gases from furnace to smokestack will be reduced to such an extent that most of the heat will go up the smokestack. In the latter case, spiral baffles placed in the tubes will improve matters somewhat. Vertical boilers answer very well as donkey boilers in moderate-sized vessels. They occupy very little floor space, and it is usually not a difficult matter to find a place for them in almost any vessel, either in the fireroom or on deck. As donkey boilers are used only when there is no steam on the main boilers, their duration of service is usually short; therefore, the little extra coal they burn does not amount to much. Vertical boilers that are intended to be used as donkey boilers on board ship need not be built as low as those for launches; consequently, the tubes may be longer, by which a higher efficiency is obtained.

There are two types of vertical boilers, designated as the *flush-tube* boiler and the *submerged-tube* boiler.

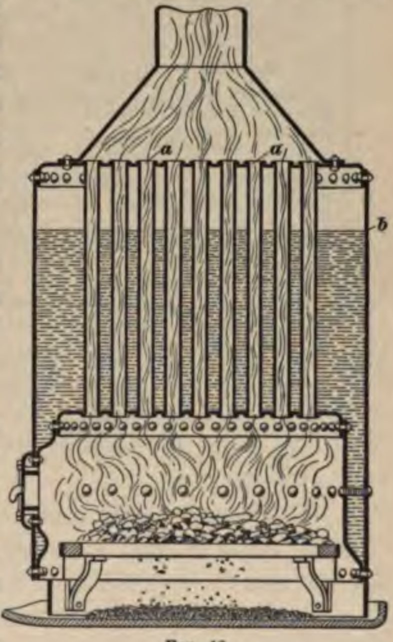


FIG. 12

18. Construction.—A vertical section of a **flush-tube boiler** is shown in Fig. 12. It will be observed that the upper ends of the tubes *a*, *a* project above the water-line *b* and are surrounded by steam, while all the remaining parts of the tubes are surrounded by water. If the fire is forced, there is some danger of overheating the upper ends of the tubes, but under moderate steaming they have a tendency to dry the steam.

19. A **submerged-tube boiler** is illustrated in Fig. 13. The shell *a*, the firebox *b*, and the uptake *c* are shown in section. The water-line is about at the center of the uptake; therefore, all the tubes *d* are entirely covered by water. This obviates the danger of burning the upper ends of the tubes. The firebox and uptake are secured to the shell by a large number of socket staybolts *e*, *e*, etc., and the heads are braced by the diagonal braces *f*, *f*. Handholes *g*, *g* are provided for the purpose of cleaning mud and sediment out of the water space of the boiler, but the tubes are not readily accessible for scaling. Therefore, pure feedwater should be used in these boilers.

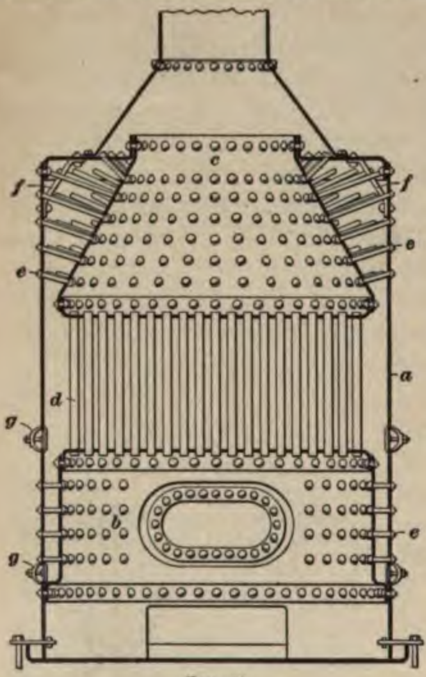


FIG. 13

WATER-TUBE BOILERS

CLASSIFICATION

20. There is such a variety of water-tube boilers on the market that it is difficult to rigidly classify them. Nearly every conceivable aggregation and arrangement of tubes and pipes have been employed to produce water-tube boilers; thus, considerably over two hundred styles of this type of boiler have been approved up to 1903 by the Board of United States Supervising Inspectors of Steam Vessels, and authorized to be installed on vessels plying on United States waters.

In general, the different varieties of water-tube boilers are designated by the size and arrangement of their tubes or pipes, namely: straight-tube and bent-tube boilers; large-tube and small-tube boilers; horizontal-tube, vertical-tube, and inclined-tube boilers; pipe and coil boilers. There is also another designation known as sectional and non-sectional boilers. There are also various combinations of the above-mentioned types forming other types, but they all follow the same general principle of the water circulating through the tubes and the products of combustion surrounding them. For the sake of convenience, the boilers described have been divided into straight-tube, bent-tube, and sectional pipe boilers. The boilers illustrated have been selected as illustrating to the best advantage the salient features of each type; the fact of their being described is not intended to prove them superior to others on the market, nor is the omission of other boilers to be construed as a mark of inferiority on their part.

STRAIGHT-TUBE BOILERS

21. Babcock & Wilcox Boiler.—A boiler of the large straight-tube type is illustrated in Fig. 14. From the name of the firm manufacturing it, it is known as the **Babcock & Wilcox marine boiler.** As illustrated, the principal pressure parts are the steam and water drum *a*, the front headers *b*, the rear headers *c*, the generating tubes *d*, *d*, the horizontal connecting tubes *e*, and the cross-boxes *f*, *g*. The upper ends of the front headers are connected to the steam and water drum by nipples, one of which is shown at *h*, and the lower ends are connected with the forged steel cross-box *g* by short nipples, one of which is shown at *i*. The upper ends of the rear headers are connected to the forged-steel cross-box *f* by nipples. The tubes *d*, *d* forming the heating surface are arranged in vertical sections, and, to insure a continuous circulation in one direction, are placed on an inclination of 15° with the horizontal. As shown in Fig. 15, the two forged-steel headers *a*, *b* and the tubes *c*, *c* connecting the headers together constitute one section. A number of these sections

placed side by side form a boiler. The sections are in communication with one another through the cross-boxes *f*, *g*, Fig. 14, and the drum *a*. The tubes of each section are expanded at their ends into the headers, which, being sinuous in form, permit the tubes to be placed staggered. The currents of hot gases are therefore completely broken up in their

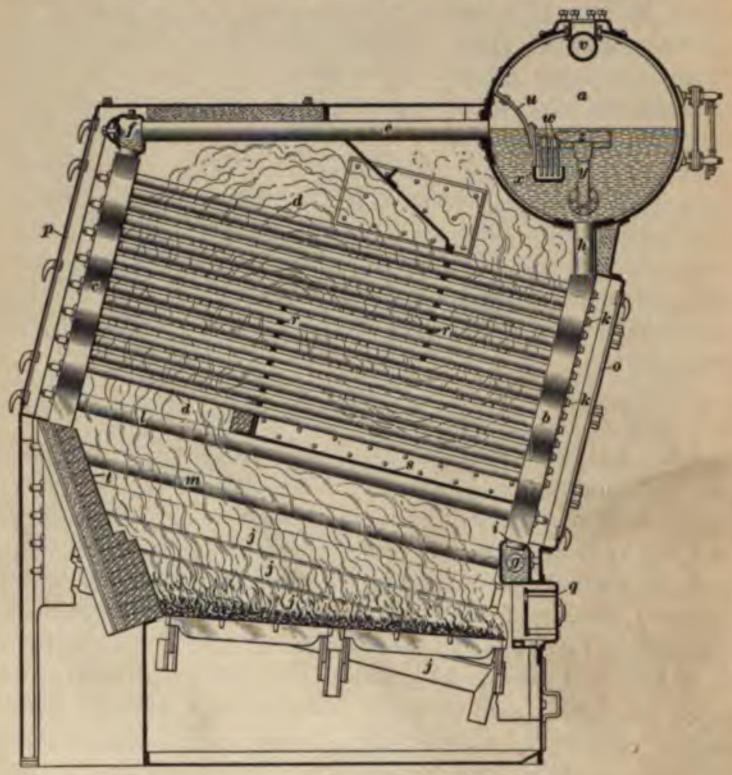


FIG. 14

passage across the heating surface, and the too free escape of the products of combustion is also prevented. By dividing the heating surface into sectional elements, the injurious effects of unequal expansion due to raising steam quickly, or unequal contraction caused by sudden cooling, are obviated, each element being thus free to expand and contract

independently of the others. The side sections are continued down to the level of the grate, the tubes being replaced by forged-steel boxes of square cross-section at the sides of the furnaces. These boxes are placed one above the other at the same angle as the tubes; they take the place of brickwork, insure a cool side casing, prevent the adherence of clinkers, and are of sufficient thickness to withstand the wear and tear of the firing tools. The square boxes are technically called *water-tube sides*, and are shown at *j, j*, Fig. 14. As mentioned before, the upper end of each front header is directly connected with the drum, while the upper end of each rear header is first connected to the cross-box *f* and then to the drum by a horizontal tube *e*; hence, each section is provided with an independent inlet and outlet for water and steam.

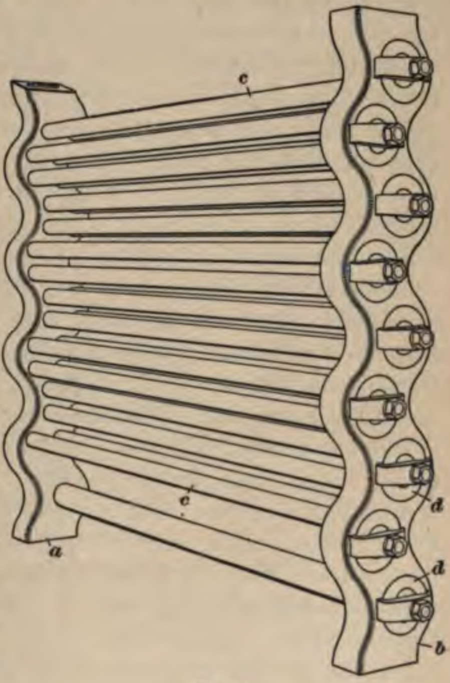


Fig. 15

The cross-box *g* is located at the lowest part of the bank of tubes and forms a mud-drum for the collection of sediment; the boiler may also be drained through it. All tubes are constructed of seamless steel and are extra heavy. The generating tubes *d, d* are 2 inches outside diameter. Opposite the ends of the tubes are openings, or handholes *d, d*, Fig. 15, in the headers through which the tubes may be examined, cleaned, plugged, or renewed. In some cases, the handholes are 4 inches in diameter, and they are closed by forged-steel plates into

which are riveted 1-inch studs. These plates are faced, and are drawn to faced seats by forged-steel bridges and nuts, the joints being made on the inside of the header, by means of thin gaskets. When the 4-inch handhole is adopted, a group of four of the 2-inch generating tubes may be taken care of through one handhole. In other cases, as shown at *k*, *k*, Fig. 14, each tube is provided with a separate opening in the header just large enough in the clear for the tube to be passed through. These holes are closed with screw plugs. Should a tube be found defective, it may be renewed or plugged, as both ends are accessible.

The lower row of the bank of generating tubes generally consists of 4-inch tubes *l* instead of groups of four 2-inch tubes. The upper tubes *m* of the water-tube furnace sides are also 4-inch. The boiler is enclosed in a sheet-metal casing lined with some refractory non-heat-conducting substance such as asbestos, magnesia, etc. Openings are made in the casing for the front and rear tube doors *o* and *p*, Fig. 14, respectively, which provide access to the ends of the tubes. One of the furnace doors is shown at *q*, in this case underneath the lower end of the bank of tubes and cross-box *g*.

The location of the steam and water drum is at the front of the boiler immediately overhead. Vertical baffles *r*, *r*, in connection with the roof *s* of light fire-tiles placed on the lower row of tubes, compel the gases to follow the circuitous route shown, crossing the heating surface three times before their exit, thus causing them to impart the greatest possible amount of their heat to the water in the tubes.

As the furnace increases in height as it approaches the firebrick bridge wall *t*, a combustion chamber of ample size is thereby provided, and the gases have both space and time in which to thoroughly mix and burn before entering the spaces between the tubes. The gases evolved from the combustion of the fuel are compelled to flow toward the rear of the furnace by the roof *s* of fire-tiles, which causes them to pass over the incandescent bed of coal and under the hot tile roof. By this arrangement, a high furnace temperature is maintained, which is an essential requirement of boiler

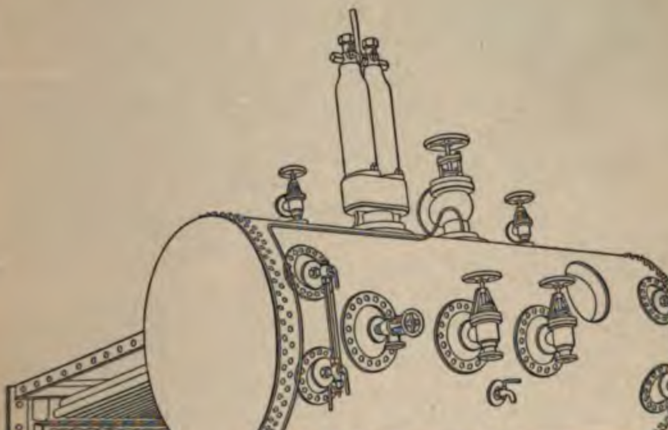


FIG. 16

economy. The distance traveled by the products of combustion in contact with the heating surface is about 16 feet; hence, good economy is maintained with high rates of combustion, and a low uptake temperature is assured.

The location of the drum in front of the boiler renders all valves and fittings accessible and tends to shorten steam-pipe connections. The main stop-valve and safety valves, feed stop-valve and feed check-valve for both main and auxiliary feeds, and the glass water gauges are flanged directly to nozzles provided with counterbored seats and fastened to the drum shell or heads.

A perspective exterior view of a Babcock & Wilcox boiler, with one pair of tube doors and part of the side casing removed, is given in Fig. 16. The boiler here illustrated differs in some minor details of construction from that shown in Fig. 14, but serves to show the general appearance.

22. The operation of the boiler is as follows: The boiler being filled with water until the steam and water drum is half full, the fires are then started. The water in the inclined tubes becomes heated first, owing to the tubes being exposed to the most intense heat of the fire; it then expands and flows up to and through the rear headers and through the horizontal tubes into the drum. Cooler water flows from the bottom of the drum into the front headers and also into the cross-box *g*, Fig. 14, which acts as the mud-drum. This water, in turn, becomes heated and flows up the inclined tubes as before, circulation being thus maintained. On entering the drum, the steam and circulating water are directed against the baffle plate *u*, which causes the water to be thrown downwards, while the steam is liberated from the water and passes around the ends of the baffle plate to the steam space. To insure dry steam, a dry pipe *v* is fitted inside of the drum; it is suspended in the upper part of the drum by hangers. The main and auxiliary steam pipes take steam through this dry pipe.

Zinc slabs *w*, *w* are suspended in the steam and water drum just below the water level to arrest corrosion caused

by galvanic action. The pan x is placed underneath the slabs to catch the pieces of zinc that fall from the slabs when they disintegrate by the action of the galvanic current.

The surface blow-off pipe is attached to the lower part of one of the drum heads; it extends inwards for a short distance and then bends upwards, as shown at y, where it terminates in the scum pan z, which collects the scum and other foreign matter floating on the surface of the water and allows it to be blown out through the surface blow-off pipe.

23. See Boiler.—A boiler of the straight inclined-tube type, which differs considerably from the Babcock & Wilcox boiler, is shown in Fig. 17. It is known, from the name of its designer, as the **See boiler.** The boiler consists of the steam drum a, the two water or mud-drums b, b and the two nests of tubes c, c. This illustration shows only the pressure

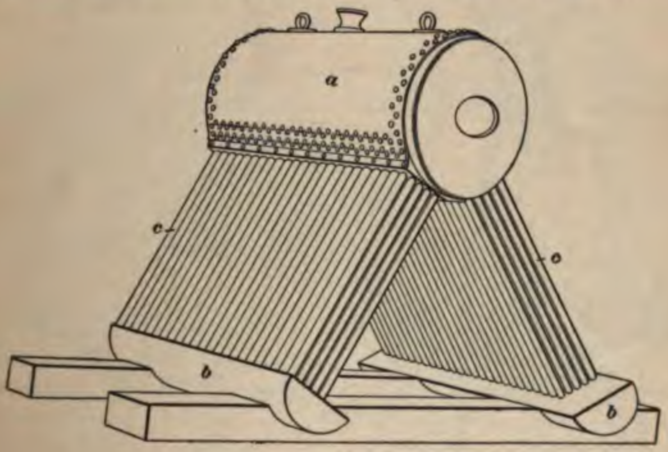


FIG. 17

parts without the casing. Like all boilers of this class, it has a large heating surface and makes steam rapidly. The tubes, being straight and accessible, can be cleaned of scale—a most important feature in any boiler.

A sectional view of the mud-drum, showing its method of construction, is given in Fig. 18, and an exterior view of the

complete boiler is given in Fig. 19. An important feature of this boiler is that the weight of the steam-generating part, consisting of the drums and tubes, is supported from the steam drum and not on the mud-drums, as is the usual custom with boilers of this type. This permits the tubes to expand and contract freely without bringing undue strain on them. The mud-drums are kept from spreading by the hinged brackets *a*, *a*, Fig. 19. The

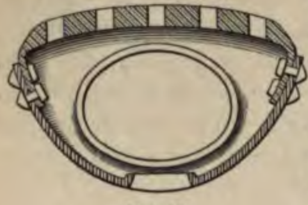


FIG. 18

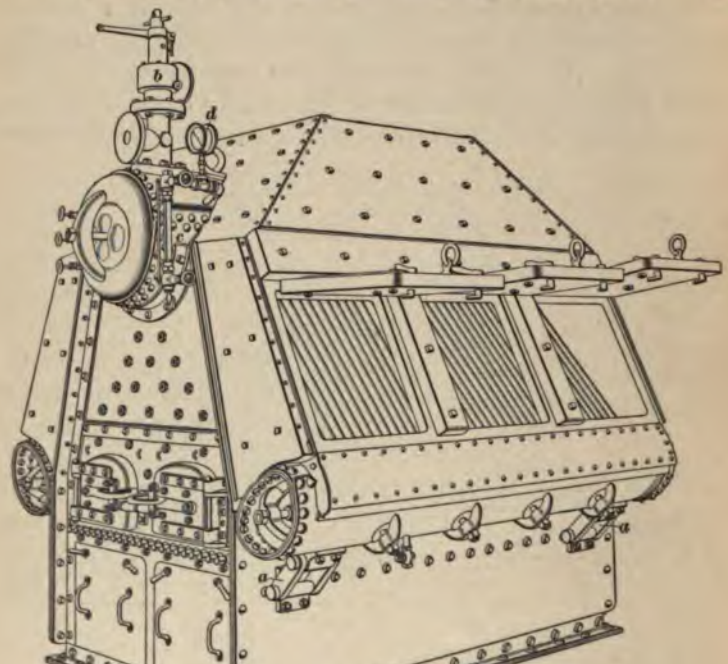


FIG. 19

safety valve *b*, gauge glass *c*, and steam gauge *d* are attached to the front end of the steam drum.

The operation of the boiler is as follows: It is filled with water to about the middle of the steam drum, and the fire is started. The inside tubes of each nest being most exposed to the intense heat of the fire, the water in them expands and rises to the steam drum. The outside tubes being located in a relatively cool place, water flows down them to the mud-drums to take the place of that flowing up the inside tubes. A steady and rapid circulation of the water is thus established.

BENT-TUBE BOILERS

24. Seabury Boiler.—The boiler shown in Fig. 20 is an example of the bent-tube type, and belongs to the class having but one bend in each tube, as shown. It is known as the **Seabury boiler,** and is applicable to yachts, torpedo boats, and other small craft. This boiler has large grate and heating surfaces for its size and weight, and makes steam rapidly when clean.

Its essential features are the steam drum *a*, the water drums *b, b*, and the tubes *c*. In common with all boilers of this class, the tubes connect the steam drums and water drums together and afford passages for the water to circulate. The water drums are semicircular in section, the tube-sheet *a*, Fig. 21, forming the flat side of the drum. The curved part *b* of the drum is made by welding a head in each end of a piece of lap-welded iron or steel pipe, dividing it into halves through the center lengthwise and planing the edges *c, c* to fit the grooves *e, e* in the under side of the tube-sheet. The joint is made tight by placing asbestos gaskets in the grooves *e, e*. The curved part of the drum is held in place by straps *f* and screw bolts *g*. The heads of the bolts are countersunk on the top side of the tube sheet, which makes a flush surface, so that ashes, etc. may be easily cleaned off. The bolts are provided with cap nuts *h, h* that protect the threads from corrosion. The straps and bolts are also shown at *d* and *e*, Fig. 20. The boiler is supported by the tube-sheets of the water drums resting on the angle bars *j, j*, which are bolted to the bunker bulkheads *k, k*,

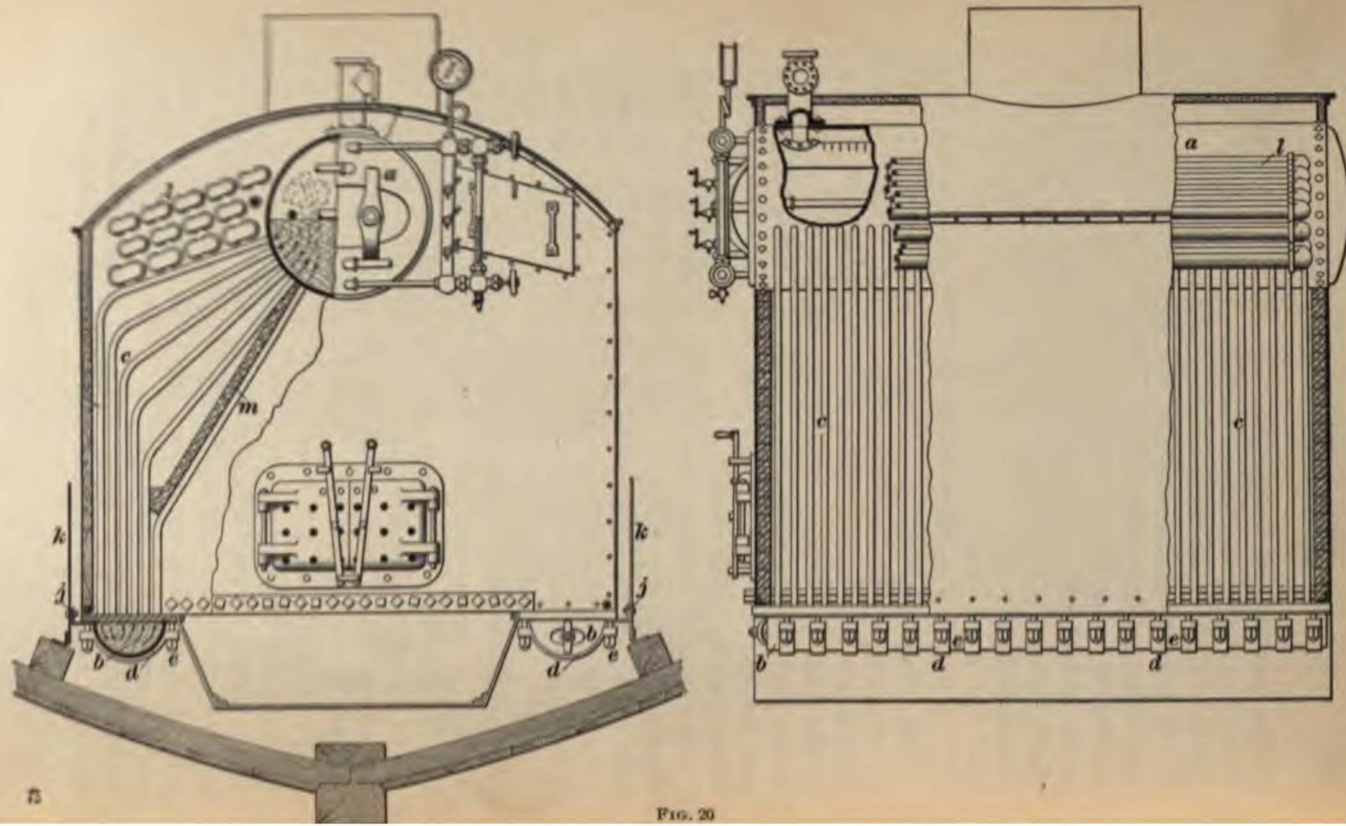


FIG. 20

Fig. 20. The tubes are expanded into the steam drum and into the tube-sheets of the water drums. The front head of the steam drum is provided with a manhole to give access to the interior for examination, cleaning, and repairs. In the small sizes of boilers, the steam drums are made of steel pipe, and the heads are bolted on at each end. Each water drum has a handhole in its front head. A feedwater heater or economizer is located in the uptake, through which heater the feedwater passes before entering the steam-generating part of the boiler, whereby considerable heat is saved that

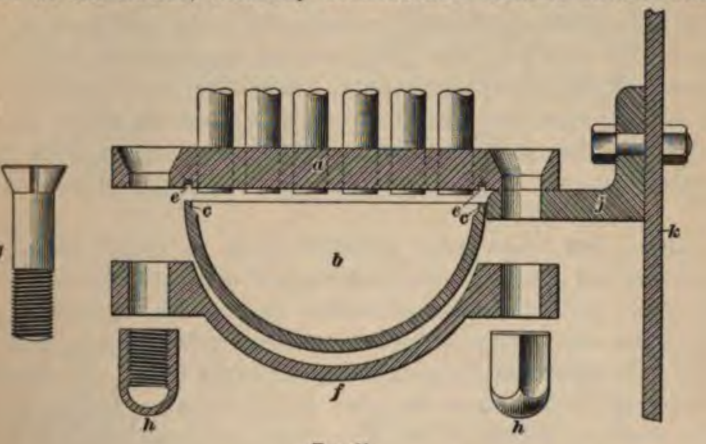


FIG. 21

would otherwise be lost through the smokestack. This apparatus consists of a bank of lap-welded iron pipe and and malleable-iron return bends. It is shown at *l*, Fig. 20. Firebrick baffle plates, one of which is shown at *m*, are placed over the inner row of tubes for the purpose of directing the flow of the hot gases of combustion amongst the lower ends of the tubes. When the fire is burning fiercely, the firebricks become very hot, and the heat thus absorbed is given off again when the furnace is cooled down by putting fresh coal on the fire.

25. The circulation of the water in this boiler is accomplished in the following manner: The boiler being filled with water up to the center of the steam drum, the fire is

lighted. The hot gases of combustion will come in contact with the inner row of tubes first, which will cause in those tubes an upward movement of the water toward the steam drum. As the heat increases, the water in the other tubes will be affected in a similar manner and will also rise to the steam drum. By the time the hot gases have reached the outer row of tubes, they have surrendered a large part of their heat to the water in those tubes with which they have been in contact; consequently, the outer row of tubes will be much cooler than the others. The rapid upward movement of the mixed water and steam in the hotter tubes causes the solid water to flow into the lower end of those tubes from the water drums, and, necessarily, the cooler water must flow down through the outer row of tubes from the steam drum to supply the place of that which has been taken from the water drums. Thus, a continuous circulation is maintained.

26. The tubes have sufficient bend to enable them to expand and contract without distortion or straining the joints at their ends. Perforated steam pipes, with the openings opposite the spaces between the tubes and pointing across the tube-sheets and upwards amongst the tubes, are placed between the outer row of tubes and the casing for blowing the soot and ashes off the tube-sheets and lower ends of the tubes. These can be put into operation at any time, even while steaming. The feedwater heater and the upper parts of the tubes can be cleaned of soot by inserting the perforated nozzle of a steam hose through properly located doors provided for this purpose in the casing. This is a very valuable feature, as it renders it possible to keep the heating surface clean at all times. Although it is possible to clean the insides of the tubes by removing the lower parts of the water drums and passing a chain with a wire brush attached through each tube from the steam drum, it is by no means an easy operation and it is hence preferable to use pure feedwater in this boiler and also in all other boilers of the bent-tube type, and thereby keep the impurities out of them entirely.

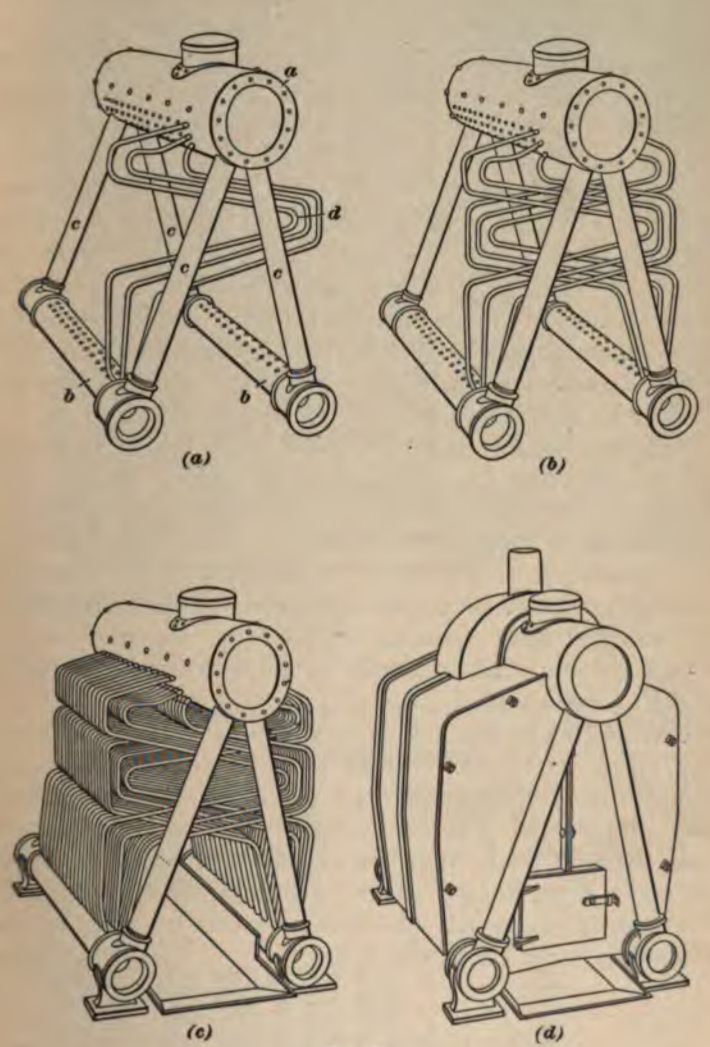


FIG. 22

27. Mississippi Boiler.—A boiler of the bent-tube type in which each tube has several bends, and belonging to a class of which there are numerous examples in existence, is the **Mississippi boiler,** shown in Fig. 22. In order to show the construction clearly, the boiler is shown in several stages of its erection, Fig. 22 (*a*) showing one set of generating tubes in position; Fig. 22 (*b*), two sets; Fig. 22 (*c*), all the tubes; and Fig. 22 (*d*), the boiler completed. It is principally adapted for yachts, torpedo boats, and small craft generally, and is a rapid steam generator that will stand considerable forcing without distortion, as the tubes are so crooked to begin with that a little more bending will do no harm. Like all boilers of this class, the generating tubes connect the steam drum with the water drums and provide an upward passage for the water and steam.

Referring to Fig. 22 (*a*), *a* is the steam drum, *b*, *b* are the water drums, and *c*, *c* are the downtake pipes for returning the water from the steam drums to the water drums, thereby maintaining a constant circulation. These elements form the framework of the boiler. The ends of the tubes are secured to the steam drums and water drums by right-and-left threaded steel bushings, as it will be observed that the drums are too small for a man to enter to expand the tubes in the ordinary way.

By referring to Fig. 22 (*d*), it will be seen that the down-flow pipes are located outside of the casing, and hence are comparatively cool, as they are not exposed to the radiant heat of the fire. The circulation of the water is upwards from the mud-drums in all the generating tubes; water from the steam drum flows downwards through the large and cool down-flow pipes to the mud-drums to take the place of that ascending in the generating tubes.

SECTIONAL PIPE BOILERS

28. Roberts Boiler.—Boilers made up of pipe and the usual forms of pipe fittings, with all or most joints made with screw threads, are generally classified as pipe boilers

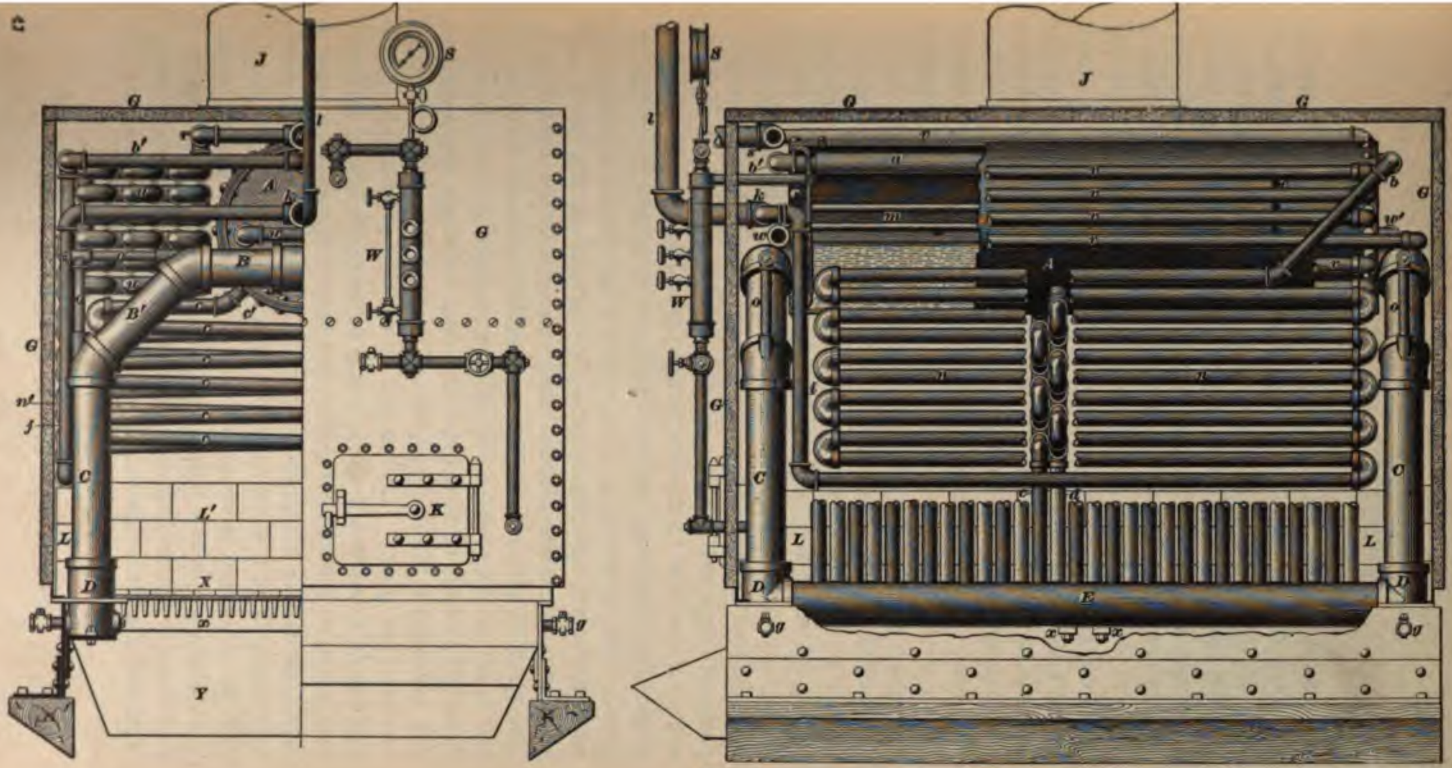


FIG. 23

and are usually sectional, that is, the generating tubes are in separate sets or sections. The **Roberts boiler**, shown in Fig. 23, belongs to this class, and has a combination of vertical and slightly inclined horizontal generating tubes. Although the tubes are straight, it cannot be classed as a strictly straight-tube boiler, because the straight sections of pipe are connected together by elbows, thus preventing access to the interior of the tubes for cleaning; hence, pure water only should be used in this boiler, as otherwise the tubes will soon become coated with scale, greatly diminishing its efficiency and shortening its period of usefulness.

The construction of the boiler is as follows: A cylindrical steam drum *A* forms a receptacle for a small body of water, the space above the water forming the steam space; it is made of a steel or iron plate, and closed by flanged heads riveted to the shell, having the heads stayed by the stayrods shown at *m*. Two side pipes *E* (only one is shown in the figure) are connected to the front and back of the steam drum by means of the down-flow pipes *C*, *C*, the angular down-flow pipes *B'*, and the cross-pipes *B*. It will be noticed that on each of the **T**'s *D*, *D* is a flange, to which are bolted the angle irons carrying the jacket *G*, *G*. A plate, on which the ends of the grate bars *X*, as well as the firebrick lining *L'*, are supported, extends across the furnace in the front and rear, the plates being bolted to the flanges of the bottom **T**'s. The grate is composed of two lengths of grate bars, which are held up at the center of the furnace by the bearing bars *x*, *x*, which in turn are supported by studs screwed into the bottom of the side pipes. The firebrick lining *L* rests on the side pipes. Connected to the top of the side pipes are the up-flow coils *c* and *d*; only two are shown in full, the rest being shown broken off; they start alternately from the right and left side pipes. These coils are composed of pipes and return bends, with the return bends tapped "on a spread"; that is, so as to give all the pipes an upward inclination. This is shown in the front elevation, which also shows at *c'* the point at which the up-flow coil *c* enters the steam drum. The coil *d* enters the steam drum at a similar point at the

opposite side. Two feed-coils u, v are placed one on each side of the drum, and are supported on the small pipes o, o. Superheating coils n, n' are placed one on each side of the furnace, outside of the up-flow coils. The boiler is enclosed at the four sides and the top by a sheet-iron jacket G, G, lined with some non-heat-conducting material. An opening is provided in the top of the jacket, through which the smoke-stack J connects with the space inside the jacket forming the combustion chamber, the space directly over the grate X being the furnace. At W, the water column, mounted with a glass water gauge and three gauge-cocks, is shown. The steam gauge is shown at S, the furnace door at K, and the ash-pit at Y.

The feedwater enters at s, and is divided into two streams by a partition cast with the **T**'s. One stream passes through the pipe r into the feed-coil u; the second stream passes through a similar pipe into the feed-coil v. The feedwater is heated to a high temperature before leaving the coils. On leaving them, the water passing through the coil u enters the steam drum at w', and the water passing through v enters at w. In operation, the entering feedwater is discharged above the water-line to allow the steam that may have formed in the feed-coils to rise to the top of the drum, and the water to fall to the water level. The horizontal layers of the feed-coils are separated by cross-pipes e, which are placed there to prevent sagging of the free ends of each layer of pipe composing the coils. In order to show the coil u more clearly, these cross-pipes have been omitted in the front view. There is no water in these cross-pipes.

29. The boiler being filled with water until the steam drum is about one-quarter full vertically, and the fire started, the water expands and becomes lighter much faster in the up-flow coils c, d than in the down-flow pipes C, C; this is due both to the fact of the small pipes absorbing a greater amount of heat, owing to their being directly over the fire, and also to their being of a smaller diameter, because there is more heating surface for a given volume of water in the

pipes. The result is that the water at once commences to rise in the small up-flow coils and to fall in the large down-flow pipes. As soon as steam bubbles commence to form, this movement of the water becomes more rapid, as water holding these bubbles is much lighter than solid water, even if their temperatures are the same. To assist the upward flow of the water, the pipes *c*, *d* are given an upward inclination.

The up-flow coils now begin to throw currents of water mixed with steam bubbles into the drum. The steam bubbles break, the steam rises to the top of the drum, and the solid water flows out of each end into the cross-pipes leading to the down-flow pipes. The lower part of the bottom **T** *D* of each down-flow pipe forms a mud, or sediment, pocket; that is, it provides a quiet place for the deposit of foreign matter held in suspension in the feedwater. This deposit may be drawn off by means of the cocks *g*, *g*. Just above the mud-pockets, the downward currents make a turn at right angles into the side pipes, part of the water flowing into the up-flow coils as it passes along underneath them. The two currents, one from each end of each side pipe, meet in the center of the side pipes and there form an eddy, the only non-circulating water in the boiler excepting that in the mud-pockets. The steam at the top of the drum passes into a spray pipe, or dry pipe, *a*, running from one head to the other, and drilled full of small holes on top for about half its length in the center. By locating the holes near the center, no water can enter the dry pipe when the vessel is pitching. The superheating coil *n'* is connected to one end of the dry pipe by the pipe *b'*, and the superheating coil *n* to the other end by the pipe *b*. The steam flows into the top of the coils and passes downwards to the bottom, rising upwards again within the pipes *i*, *j* that unite at *k*, whence the steam passes into the pipe *l*. To the upper end of this pipe a **T** is attached, to one outlet of which the safety valve is fitted, while to the other outlet the steam pipe leading the steam to the engine is connected. These connections, for want of room, are not shown. The superheating coils *n*, *n'* answer the double purpose of drying the steam and of protecting the jacket from

the action of the fire. The object of the angular down-flow pipes B' is to prevent any part of the down-flow system (the pipes B, B', C) from being thrown above the water-line by a listing of the vessel, which would greatly interfere with the circulation.

30. The Roberts boiler is an example of one of the two general types into which water-tube boilers are sometimes divided, viz., **drowned-tube** boilers, or boilers in which the upper ends of the tubes are submerged in water, and **dry-tube** boilers, or boilers in which the upper ends of the tubes are filled with steam only, or such water as is carried up by the violent ebullition of the water within the lower part of the tubes. From the construction of the Roberts boiler, it follows that it belongs to the drowned-tube type.

31. Almy Boiler.—An example of the dry-tube sectional-pipe boiler is shown in Fig. 24. It is composed of straight pipes, part of them being nearly horizontal and part of them vertical, and connected together by elbows, return bends, and **Y** fittings. From its designer, it is known as the **Almy boiler.** The interior of the tubes are inaccessible for cleaning; hence, pure feedwater only should be used in it, as well as in all pipe and bent-tube boilers.

The construction of the boiler is as follows: A continuous manifold c, which forms the base of the boiler, extends along the sides and across the back of the boiler. At the top is a similar manifold a, which extends along the sides and across the front of the boiler. To form the heating surface, a series of up-flow coils D, d, made up of pipes connected together by elbows, return bends, and four-way **Y** fittings, are connected to the top and bottom manifolds and with each other by means of unions. The coils D at the sides of the boiler rise from the manifold to a proper height to form the top of the furnace; they then extend half way across the furnace and return to the sides; thence up, connecting to the side pipes of the top manifold a. The coils d that form the back of the furnace rise from the back manifold to a height sufficient to cross over above and at a right angle to those forming

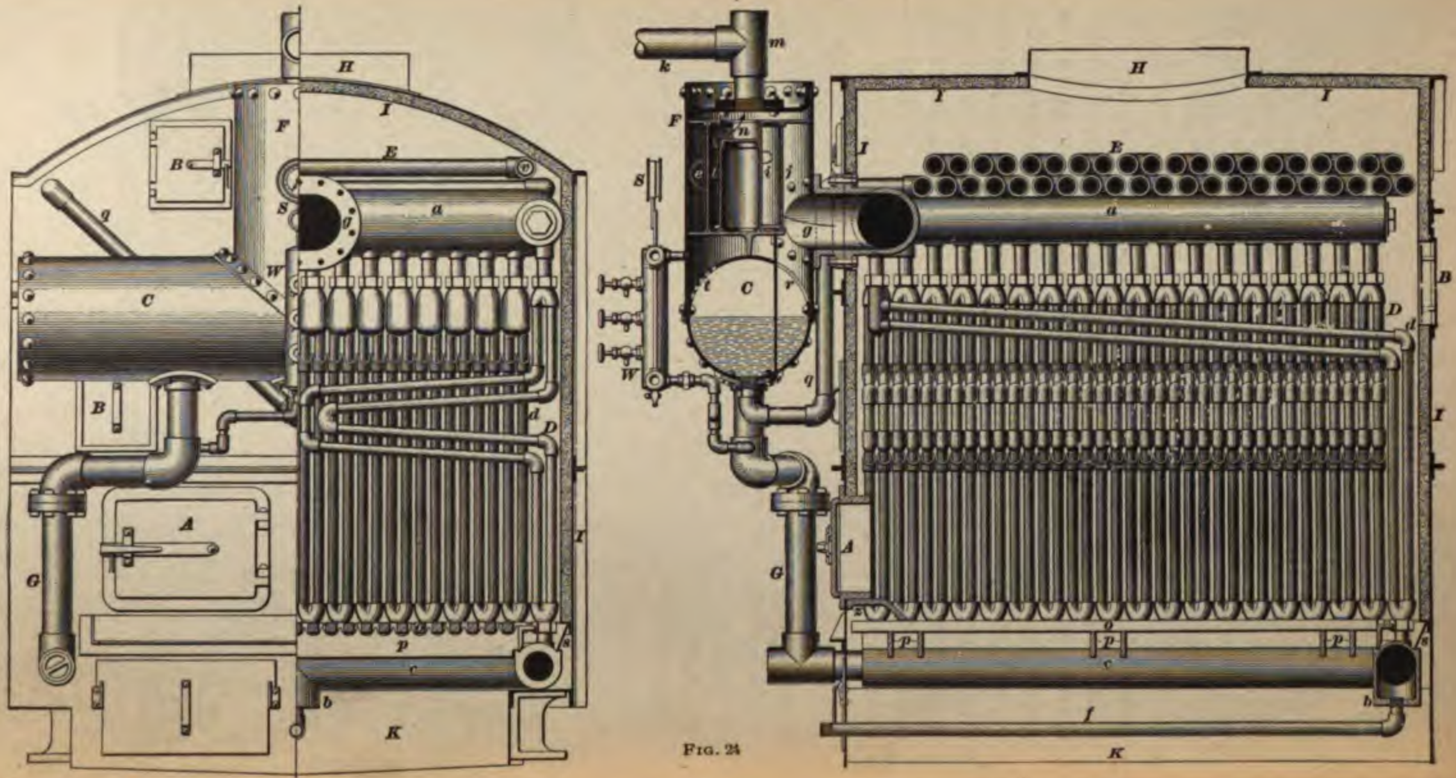


FIG. 24

the top of the furnace; they then pass to the front and connect to the manifold extending across the front at the top. A feed-coil *E*, serving to heat the feedwater, and consisting of two layers of pipe connected together by return bends, thus forming a continuous pipe, rests on the top manifold. The feedwater enters this coil at *v*, and after passing through, leaves it by means of the pipe *q* connected to the bottom of the horizontal water reservoir *C*, which extends across the front of the boiler. The top manifold *a* is attached by means of the nozzle *g* to the vertical separator *F* placed on top of the horizontal water reservoir. A jacket *I*, *I*, lined on the inside with some non-heat-conducting material, encloses the boiler on the four sides and at the top. To the ring *H*, on top of the jacket, the smokestack is attached. The grate is composed of square iron bars *o*, *o* that are supported by the bearing bars *p*, *p*. To prevent any entrance of air to the furnace, otherwise than through the grate, baffle plates *s* extend around the sides and back of the furnace. At the front, the dead plate *z* serves the same purpose. The boiler is provided with a water column *W*, mounted with a glass water gauge and three gauge-cocks, and which also carries the steam gauge *S*. At *A*, the furnace door is shown. *B*, *B* are doors in the jacket to allow inspection, etc. of the inside, and *K* is the ash-pit. In the center of the bottom back manifold, a mud or sediment pocket *b* is located, whence the sediment may be drawn off or the boiler emptied through the pipe *f*. Both the separator and the water reservoir, as well as the down-flow pipes *G*, *G*, which are connected to the water reservoir and extend down to the manifold at each side of the furnace, are outside the jacket, and hence in a cool place.

32. The operation of the boiler is as follows: The boiler being filled with water until the reservoir *C* is about half full vertically, the fire is started. The water in the coils *D*, *d* expanding, and hence becoming lighter, rises to the top manifold, carrying the steam bubbles with it; cooler water, coming from the reservoir *C* through the down-flow pipes *G*, constantly takes its place, and in turn becomes heated and

rises. On reaching the top manifold, the steam bubbles burst, the steam and entrained water rising to the top; the water thrown into the top manifold by the violent ebullition within the tubes, after flowing along the bottom, falls through the opening *r* in the horizontal water reservoir to the water level. The steam and entrained water pass into the separator *F*, where, by curved partitions, the steam is constrained to move in a spiral path; that is, it passes from *g* into *j*; thence into *e*; thence into the passage *i*, whence it passes into *l* and out of the separator at *n*, entering there the bottom of the steam pipe *k*. By giving the steam a whirling motion, the particles of entrained water are, by the action of centrifugal force, thrown against the curved partitions; the water drips down these partitions and flows back into the reservoir *C* through an opening in the base of the separator (not shown in the figure) and through the holes shown at *t*. To the top of the T shown at *m*, the safety valve is attached.

COMPARISONS

ADVANTAGES AND DISADVANTAGES OF SCOTCH BOILERS

33. Until recent years, the Scotch boiler has been installed almost universally in ocean-going steam vessels of the world, both in the merchant service and in the various navies. For numerous reasons, it has been the most efficient marine boiler in use up to a recent period, and, although it has been superseded by the water-tube boiler on the larger high-speed vessels of the merchant service and navies, it still continues to be the favorite boiler for other vessels. The principal meritorious features of this boiler are as follows: It is durable under rough usage, and easy to take care of and repair. The tubes being straight and of standard sizes, they can be procured in any port of commercial consequence in the world. Being straight, the tubes can easily be cleaned of soot, and while steaming if necessary. A leaky tube can be plugged without blowing off the pressure

from the boiler, and a new tube can be put in easily; to do this however, the boiler, of course, must be blown down to a point below the defective tube. The evaporation results, that is, the number of pounds of water evaporated per pound of coal, are satisfactory, and experience has shown that it is no more liable to leakage than other shell boilers.

34. The Scotch boiler possesses serious disadvantages, namely:

1. *Excessive Weight.*—Thus, a modern eight-furnace double-ended boiler suitable to generate steam for a 3,300-horsepower engine will weigh, when filled, about 110 tons. A water-tube boiler of the same capacity and of a heavy type will weigh only about 80 tons; and if of light type running under forced draft, as low as 50 tons.

2. *Limit of Pressure Caused by Constructive Reasons.*—This limit, for large boilers, may be placed in the neighborhood of 200 pounds per square inch, when the shell plates become so thick as to present serious difficulties in their working and handling.

3. *Bad Effects of Unequal Expansion and Contraction.* Owing to the rapid expansion of those parts of a Scotch boiler that are directly in contact with the burning gases—notably the furnace flues and combustion chambers—and the comparatively slow expansion of the other parts, great strains occur between the hot and cooler parts that will sooner or later entirely destroy the boiler if precautions are not taken to prevent it by heating the boiler up very slowly and gradually in order to equalize the expansion as much as possible. This requires considerable time, from 4 to 10 hours, the latter, and even more time being preferable, and the same length of time should be allowed the boiler to cool down; otherwise, contraction will act on them in a similar manner to unequal expansion. This loss of time in getting up steam and cooling down is a serious detriment to vessels in which the exigencies of their service frequently demand a rapid getting under way or short stays in port.

COMPARISON OF WATER-TUBE BOILER TYPES

35. Second in importance only to having a boiler strong enough to stand the pressure of steam carried, is its accessibility for cleaning and repairs. A water-tube boiler may produce most excellent results as a steam generator when new and clean both inside and outside, but if it cannot be cleaned and repaired its high efficiency will be of short duration. Unless pure feedwater is used, scale will soon form on the inside of the tubes, which will retard the heat of combustion entering the water, and some of it will go up the smokestack instead. Thick scale on the inside of a tube and a coating of soot on the outside renders the tube nearly valueless as a heat transmitter, and if these deposits cannot be removed, the boiler will in a short time become of very little use as a steam generator.

The claim is made that the circulation of water in a water-tube boiler is so rapid that the impurities in the water are swept through the tubes so quickly that they have not time to adhere to the tubes. While this is true so far as the mud and other foreign matter held in mechanical suspension in the water is concerned, it does not hold good in regard to the impurities held in solution by the water, which are precipitated by the heat and adhere very tenaciously to the hot metal of the tubes.

Manifestly, then, the water-tube boiler that can most easily be cleaned and repaired, other things being equal, is the best boiler.

36. Straight-tube boilers possess some advantages over bent-tube boilers. The most important one is that, if the tubes are properly arranged, a scaling tool can be passed through them and the scale removed, which cannot be done with most bent-tube boilers under any circumstances. Another advantage is that it is comparatively an easy job to cut out a defective tube and replace it with a new one in a straight-tube boiler, whereas in most of those of the bent-tube type, a faulty tube, especially if it is in one of the inner

rows of a nest of tubes, as it is very apt to be, is very difficult of access, and it is usually necessary to cut out a number of sound tubes to reach the one that is defective. Again, the straight tube provides a more direct passage for the circulation of the water and with less friction. Yet another important advantage in favor of the straight tubes is that they are of standard sizes and do not have to be bent into special shapes by special machines; hence, they can be obtained in any commercial port in the world, and as every tube will fit into any pair of holes opposite each other, they are interchangeable, and a large number of differently shaped tubes will not have to be carried, as would be the case with bent-tube boilers.

Bent-tube boilers are usually more compact and lighter than the straight-tube type for the space occupied; hence, they are almost indispensable for torpedo boats and racing yachts, where lightness and compactness are of primary importance, and a very high speed must be maintained for a limited time even if it results in the ruination of the boiler. Moreover, bent tubes are less liable to injury from excessive expansion due to the severe forcing of the fires that is occasionally necessary, or from raising steam quickly in a cold boiler, than are straight tubes. This is due to the fact that each tube can spring and take care of its own expansion independently of the others.

37. The merits and demerits of large-tube and small-tube boilers very nearly balance each other, and it is a much-mooted question among engineers as to which is the better. This matter is generally determined by the size of the boiler, however, the larger boilers being fitted with large tubes and the smaller boilers with small tubes. Tubes in marine boilers vary in diameter from 1 inch to 4 inches. Large tubes require fewer joints for a given amount of heating surface, and they may be made thicker without materially affecting their internal capacity. They are not so liable to have all the water in them suddenly converted into steam under extreme forcing conditions, and leave the tube exposed

to overheating, as might be the case with small tubes. They have the following disadvantages, however: Should a large tube be ruptured, a much larger volume of steam and water would be discharged and more damage would likely be done than if a small tube should burst. Should it be necessary to plug a large tube, a larger amount of heating surface would be rendered ineffective than would be the case if a small tube is plugged. It will also take longer to raise steam in a large-tube boiler, on account of the large volume of water in comparison with the amount of heating surface.

38. The tubes of water-tube boilers are placed at all possible angles, from horizontal to vertical; yet there is considerable difference in the efficiency of the boiler, depending on how the tubes are arranged in this respect. When the tubes are placed at an angle less than 15° from the horizontal, the steam is delivered spasmodically, in gulps, from both ends of the tubes, which produces foaming; and when the fires are forced, this action may at times leave the tubes unprotected by the water, which invites overheating of the tubes. The deposits of scale and soot will be greater on horizontal or nearly horizontal tubes than on those that are placed at a considerable angle, which renders them more liable to injury from the fierce heat of the fire. The water does not circulate as freely through horizontal tubes as through those placed at an angle, owing to the fact that the heated water and steam have a strong tendency to rise, and this tendency is resisted by the horizontal position of the tubes; consequently, the steam and water are compelled to struggle along to the ends of the tube.

There are also certain objections to placing the tubes vertically, especially if they are connected directly with the steam drum. In this case, the water will be forced out of the tops of the tubes in jets and fill the drum with a mixture of steam and water. As this action will be violent and continuous, the water will have no chance to settle to its true level and much of it will be carried into the steam main with the steam, under which conditions dry steam will be an

impossibility. This objectionable action is intensified when the tubes are short and of small diameter, and when the fires are forced; moreover, when the water is violently shot out of the tops of the tubes, they will be left dry at times and be burnt, as the water will not be able to flow into the lower ends of the tubes as fast as it is projected out of their upper ends. However, if the upper ends of the tubes are connected to cross-boxes and the cross-boxes connected with the steam drum by horizontal or nearly horizontal tubes, this bad action will, to a considerable extent, be arrested.

39. Although pipe boilers are used extensively on board yachts and similar small craft, they possess some objectionable features. Pipe boilers are usually made of ordinary lap-welded pipe and cast-iron or cast-steel fittings with screw joints—a combination not very durable when exposed to fierce heat. Lap-welded pipe is very liable to split under high pressure, and the chief object of water-tube boilers is to furnish high pressures; consequently, it would be an improvement if lap-welded pipe was discarded in these boilers and solid-drawn tubes substituted. The fittings, as stated, are made of cast metal—a material that cannot be depended on when exposed to the intense heat of the furnace. The multiplicity of screw joints exposed to the fire is another very objectionable feature, as they are sure to leak sooner or later. Boilers of this type should have the joints and fittings protected from the fire in some way, or else they should be placed outside of the heated parts of the boiler. These boilers, as a rule, cannot be cleaned of scale; therefore, it is absolutely necessary for their preservation that pure feedwater only should be used in them. It is almost impossible to repair a pipe boiler without taking it almost entirely apart, which operation will destroy a large part of it if the tubes are much worn and the screw joints are stuck fast.

40. Sectional boilers are those that are made up of sections that consist of groups of tubes put together in the shop and the sections assembled on board the vessel to

constitute a complete boiler. This is a great convenience in reboilering a vessel, as the sections can be lowered into the fireroom through the hatchways, and the necessity of cutting a large opening in the deck is obviated. For a new vessel, however, the advantage of this method of placing the boiler is not so great. A boiler can be more conveniently put together in the shop, in which there is more room to work and more light than in the hold of the vessel.

41. It is to be understood that the foregoing is merely a criticism of water-tube boilers in general. There being such a vast number of designs of these boilers, it is impracticable to deal with each individually and the points presented may not apply strictly to individual boilers of the different varieties.

ADVANTAGES AND DISADVANTAGES OF WATER-TUBE BOILERS

42. Properly designed and constructed water-tube boilers possess numerous advantages for marine purposes over shell boilers, one of the most important advantages being that a disastrous explosion cannot occur with them. The worst that can happen in the form of a rupture is the bursting of one of the tubes, which will liberate a comparatively small volume of steam and hot water without doing serious damage to the vessel. Owing to the small diameter of the tubes and other pressure parts, they are capable of sustaining with absolute safety a much higher pressure than a shell boiler. They can be forced to almost any extent, or steam may be raised in the shortest possible time, without injury, as sudden and excessive expansion of their various parts will not produce undue strains. They occupy less space, have larger areas of grate and heating surfaces for the space occupied, and are much lighter than shell boilers. This latter attribute is owing largely to the fact that they carry much less water than shell boilers, and that the elements composing their pressure parts being smaller they may be made of thinner metal. The saving in weight is an important feature in modern high-speed ocean steamers and cruisers.

A well-designed water-tube boiler is easily cleaned and repaired, all parts being generally accessible from the outside; this cannot be said, however, of all water-tube boilers, or of shell boilers either. A defective tube can be cut out and a new one inserted, or it may be plugged as a temporary expedient; and as all these operations are simple they can be performed by the regular engineer's force of the vessel.

43. There are no serious disadvantages of a properly designed water-tube boiler, except, perhaps, the cost. They are more expensive to build than shell boilers, but they are worth the difference in their greater economy of operation, smaller space occupied, and lesser weight.

MARINE-BOILER DETAILS

WATER AND STEAM SPACES

SHELL

FORMING THE SHELL

1. The **shells** of fire-tube and flue boilers and the drums of large water-tube boilers are built up of curved plates of iron or steel, technically known as **boiler plates.** These plates are a product of the rolling mill, where they are manufactured by passing very highly heated iron blooms or steel ingots between rolls operated by a powerful engine. Boiler plates vary in thickness from $\frac{3}{16}$ inch to 2 inches, the latter being about the maximum thickness that has been attained. Plates less than $\frac{3}{16}$ inch are sometimes called *sheets*, but these are not used in the construction of the pressure parts of a boiler. They are, however, used in the construction of smokestacks, front and back connections of fire-tube and flue boilers, and in the casings of water-tube boilers. Plates that are intended for marine boilers are usually sheared at the rolling mill to the sizes ordered by the boiler manufacturer.

After the arrival of the plates at the boiler works, their edges are planed, usually on a slight bevel, in a planing machine. They are then passed between large rolls, three in number, and bent to a cylindrical form. The rivet holes are then marked off and enough of them are drilled to enable the different plates to be bolted together to form the complete

shell. If butt joints are to be used, the cover-plates are prepared meanwhile; a few holes are drilled in them and they are bolted in their places, after which the rest of the holes are drilled through the cover-plates and the shell plates. After all the holes are drilled, the plates are taken apart and the burrs removed; then they are bolted together again and the riveting is commenced. The circumferential seams in the shell of a Scotch boiler are usually lap jointed and the longitudinal seams butt jointed.

ASSEMBLING THE BOILER

2. While the shell is under construction, the heads are being flanged, and, if the boiler is of the Scotch type, the combustion chambers are being erected and the furnace flues, if they are of the built-up type, are being made; but, if they are of the corrugated type, they have been ordered from the makers. After all the riveting on the shell is completed, the heads are fitted to their places, the rivet holes are marked and drilled, and the burrs removed, after taking the head out. The combustion chambers and furnace flues are next riveted to the front head, which is then placed in the boiler and riveted. Then the rear head is put in and riveted. Next, in order, are the tubes, which are inserted and expanded in the holes prepared for them in the front head and front sheet of the combustion chambers. The crown bars are now put into place on the top sheets of the combustion chambers; the diagonal braces, if any are used, are fitted to stay the heads, and finally the longitudinal braces are put into place; then, after calking the seams, the boiler is ready for its setting.

In the construction of the firebox or locomotive boiler, the firebox is built up and introduced before the front head is put on. Sling stays are also used in connection with the crown bars in this boiler to provide additional support for the flat top of the firebox.

In assembling the parts of a flue boiler, the flues are placed in their proper positions inside of the boiler and temporarily

secured there while the heads are being fitted and riveted in. In this case, the furnace is outside the boiler, it being externally fired, while the Scotch and firebox boilers are internally fired.

RIVETED JOINTS

RIVETS

3. Common forms of **rivets** are shown in Figs. 1 to 5. In Figs. 1 and 2 are shown examples of hand riveting; in Fig. 1, the head is hammered down to a cone, while in Fig. 2 the rivet has a cup, or snap, head. This form of head is produced by first hammering the rivet down roughly and then finishing the head by a cup-shaped die called the **button**

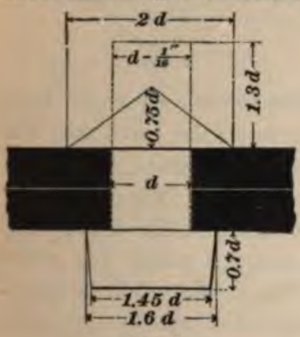


FIG. 1

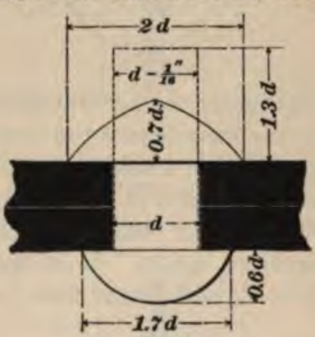


FIG. 2

set. The dotted lines in the figure show the shape of the rivet shank before being upset by the hammer. The rivets shown in Figs. 3 and 4 are examples of machine riveting. The rivet is placed between two dies that are forced together by heavy steam or hydraulic pressure. The most important advantages of machine riveting are the following: On account of the force with which the plates can be held together while the head is being formed, a tighter joint can be made; the heavy pressure used to upset the rivet and form the head causes it to expand and fill the hole more completely than it will when headed by the blows of a hammer; when a large number of rivets is to be driven,

machine riveting is cheaper than hand riveting. A rivet with countersunk head is shown in Fig. 5. Such riveting is sometimes necessary where a smooth surface is needed for the attachment of boiler mountings. In Figs. 4 and 5, the holes in the plates are countersunk slightly under the rivet heads. This provides for an increase in the size of the rivet

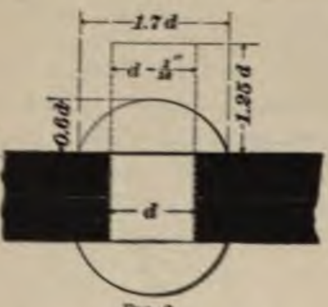


FIG. 3

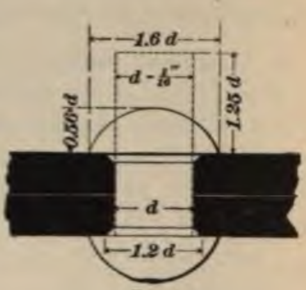


FIG. 4

just under the head and makes the rivet much stronger than is the case where the connection between the head and the rivet forms a sharp angle, as in Figs. 1, 2, and 3.

As shown in the figures, the rivets, before being headed, are slightly smaller than the hole, so that they may be inserted easily. It is the general rule to make the rivet hole $\frac{1}{16}$ inch larger than the rivet. When the work is properly done, the upsetting action of heading the rivets causes them to fill the holes when headed down. The proportions usually given to the rivet heads, and the distance the rivet shank projects from the plate before heading, are given in terms of the rivet diameter, when driven, that is, in terms of the rivet

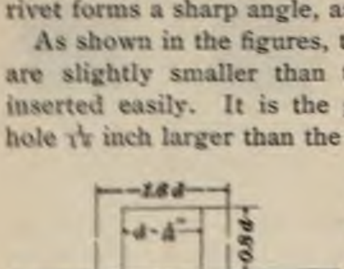


FIG. 5

hole, as shown in Figs. 1 to 5. Thus, in Fig. 1, the diameter of the upper head is given as $2d$; when d, the diameter of the rivet hole, is $1\frac{1}{16}$ inches, the diameter of the upper head will be $2 \times 1\frac{1}{16} = 2\frac{1}{8}$ inches.

4. The rules prescribed by the Board of United States Supervising Inspectors of Steam Vessels provide that all rivet holes and holes for staybolts must be drilled fairly. This rule applies to all boilers coming under their jurisdiction.

FORMS OF RIVETED JOINTS

5. Riveted joints of different forms are shown in Figs. 6 to 11. When one plate overlaps the other and the two are

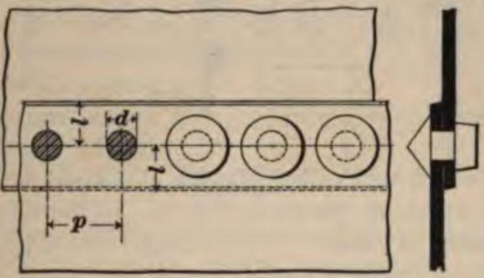


FIG. 6

joined with one or more lines of rivets, as shown in Figs. 6 and 7, the joint is said to be **lap riveted.** When, however, the plates are placed edge to edge, as in Figs. 8 and 9, and

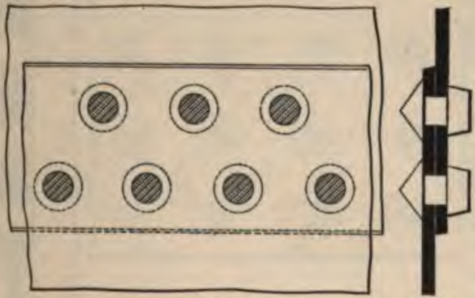


FIG. 7

the joint is covered with one or two plates, the joint is called a **butt joint.**

Fig. 6 represents a **single-riveted lap joint,** that is, the plates are overlapped and joined with one row of rivets. The distance p from center to center of the rivet holes is

called the **pitch** of the rivets. The distance *l* from the center line of the rivet hole is usually made $1\frac{1}{2}$ times the diameter *d* of the rivet hole. The distance that the two plates overlap, that is, the distance from the edge of one plate to the edge of the other plate and at the joint, is called the **lap**.

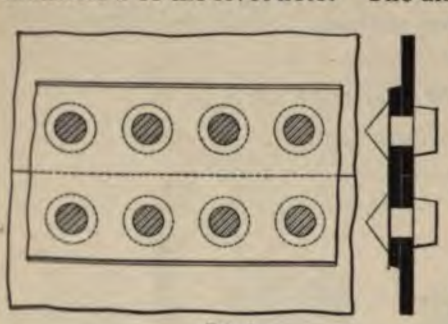


FIG. 8

Fig. 7 shows a **double-riveted lap joint.** The rivets may be **staggered,** as shown in the figure, which method is commonly called **zigzag riveting,** or placed one behind the other, as shown in Fig. 8. In the latter case the joint is **chain riveted.** In a zigzag riveted joint, the distance from the center of one rivet to the center of the next

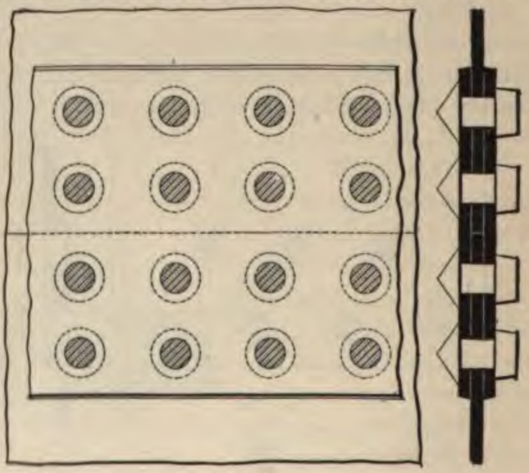


FIG. 9

rivet in the other row is called the **diagonal pitch.** It is quite customary in boiler construction to single rivet the girth seams and double rivet the longitudinal seams, since the stress on the latter is twice that on the former.

A butt joint with a single cover-plate is shown in Fig. 8, while a butt joint with two cover-plates is shown in Fig. 9. Either may be riveted with one, two, or more rows of rivets.

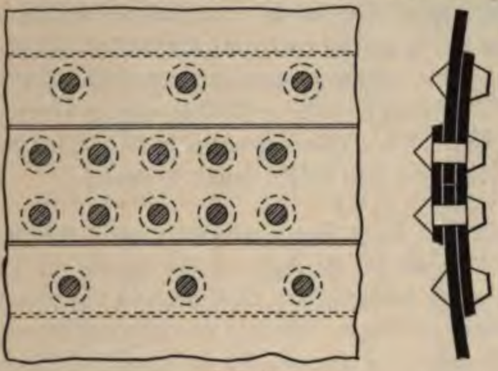


FIG. 10

Fig. 9 shows an example of chain-riveting. A well-designed butt joint with two plates will be stronger than one with a single plate. Butt joints are generally used for plates over

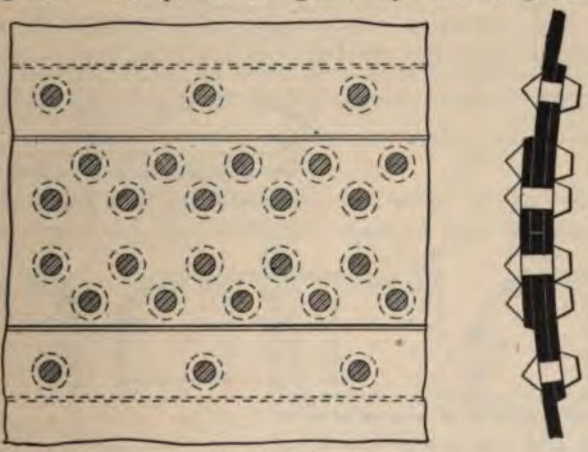


FIG. 11

$\frac{1}{2}$ inch thick and are taking the place of lap joints for longitudinal seams in good designs of smaller work. When one cover-plate is used on a butt joint, its thickness should not be less than $1\frac{1}{8}$ times the thickness of the plate; when two

cover-plates are used, the thickness of each should not be less than about five-eighths of the plate thickness.

It is sometimes the case that the cover-plates of double-strap butt joints are made unequal in width, the wider cover-plate being placed inside of the boiler. Two examples of a joint of this character are illustrated in Figs. 10 and 11. The joint shown in Fig. 10 is double zigzag riveted; the one shown in Fig. 11 is triple zigzag riveted. The cover-plates of a butt joint are also called **butt straps.**

6. Attempts have been made to weld the longitudinal seams in the shells, as well as the seams of the internal parts of Scotch boilers, but this method has not yet passed the experimental stage. Should this process be satisfactorily accomplished in the future, the labor of drilling the rivet holes, the riveting, calking, and some of the flanging will be saved.

ARRANGEMENT OF JOINTS

7. The plates of externally fired boilers should be arranged so that the riveted joints are as far as possible

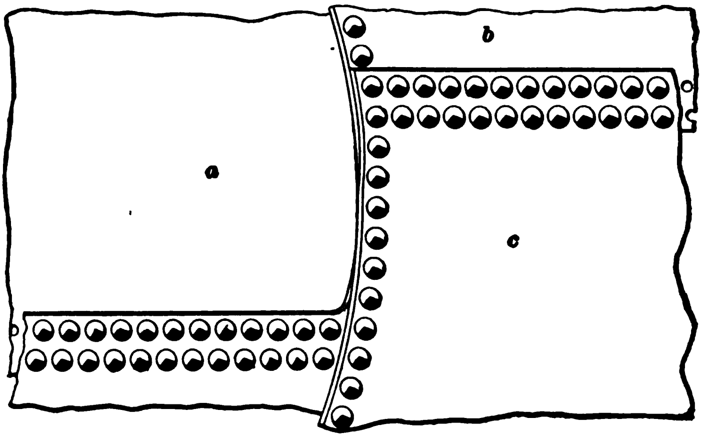


FIG. 12

from the fire. This may be accomplished by using extra large plates for the furnace end of the shell.

Wherever a girth seam occurs, the longitudinal seams should break joint, as shown in Fig. 12. In order to make a tight joint where three plates come together, the inner plate of a longitudinal lap joint must be hammered thin at the edge, as shown in Fig. 13.

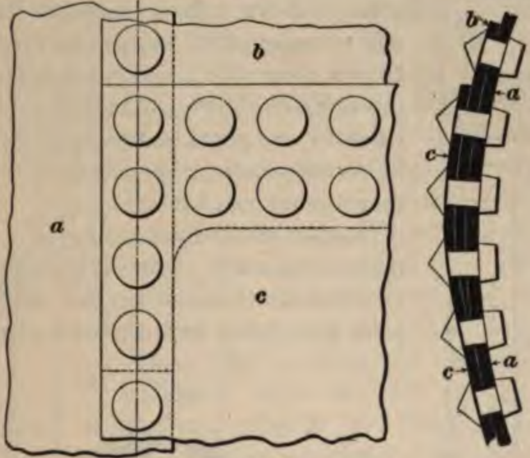


FIG. 13

In the construction of both vertical and horizontal shells, it is customary to have the inside lap facing *downwards*, since, if it faces upwards, a ledge is formed on which sediment may be deposited.

Since wrought-iron plates are stronger in the direction of the fiber, they should be arranged so that the fiber runs


FIG. 14


FIG. 15


FIG. 16


FIG. 17

circumferentially around the shell; that is, in the direction of the girth seams.

8. Different methods of connecting plates at right angles are shown in Figs. 14 to 17. In Fig. 14, the two plates are riveted to an angle iron. This construction was

formerly used for connecting the heads of a boiler to the shell, but since high steam pressures have come into use, this method has been abandoned as unsatisfactory. As shown in Figs. 15 and 16, the head is flanged and riveted to the shell, while in Fig. 17 the head and shell are connected by a flanged ring. The methods of connection shown in Figs. 15 and 16 are generally considered preferable to those that are shown in Figs. 14 and 17, since in the latter methods there are two joints to be kept tight, while in the former there is but one.

Iron or steel for flanging should be of the best quality. The radius of the curve to which the head is flanged should be at least four times the thickness of the plate.

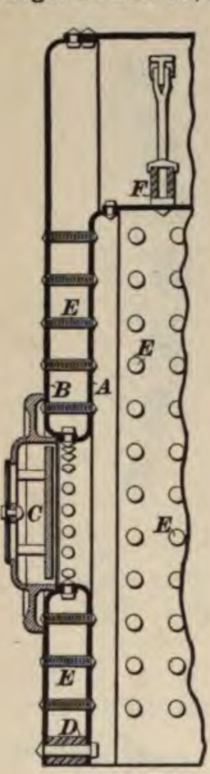


FIG. 18

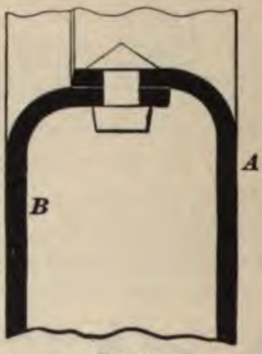


FIG. 19

Some makers of large boilers prefer to flange the end plates of the shell to receive the head, which is, consequently, a flat disk.

In Figs. 18 to 24 is shown the usual construction of the water legs and furnace doors of vertical and firebox boilers. Fig. 18 shows the door constructed by flanging the furnace sheet *A* and the front sheet *B* of the boiler. In the figure, the joint is single riveted, although it is frequently double riveted. An enlarged view of this construction is shown in

Fig. 19. The door C is generally made of cast iron and is hinged to a cast-iron frame that is usually held in position by four studs. Sometimes the frame is omitted and the door is made of wrought iron; the door is then held in position by riveting the hinges to the boiler.

Around the lower ends of the water legs, or around the bottom of the furnace, and between the inside and outside plates is riveted a wrought-iron ring D; in cheap boilers, this ring is frequently made of cast iron. Instead of flanging both sheets, as shown in Figs. 18 and 19, the furnace opening is sometimes constructed as shown in Figs. 20 and 21. A hole is cut in the outer sheet C, and the furnace sheet A is flanged. The flanged ring B is then riveted to the plates A and C, and forms the opening for the door. An enlarged view of this construction is shown in Fig. 21. A flanged ring D, Fig. 20, is sometimes used at the bottom of the

FIG. 20

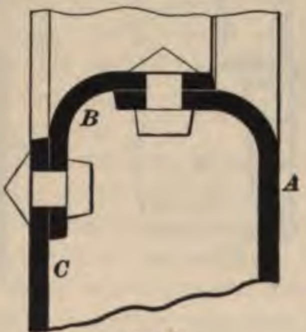


FIG. 21

water leg in place of a wrought-iron ring D, Fig. 18, one of the flanges being riveted to the furnace plate and the other to the shell, as shown. An enlarged view of this construction is shown in Fig. 22. In Fig. 23 is shown another method of constructing the opening for the furnace door and the bottom of the water leg. In this construction, the

wrought-iron ring A is placed between the furnace plate B and the shell C of the boiler, and riveted to them. An enlarged view of this construction is shown in Fig. 24. At

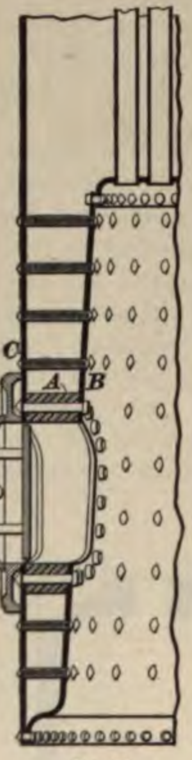


FIG. 23

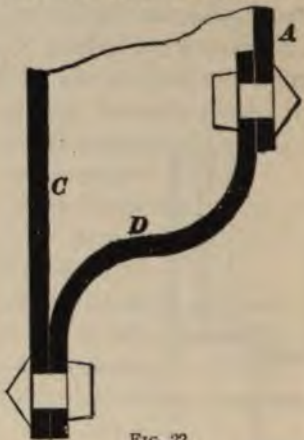


FIG. 22

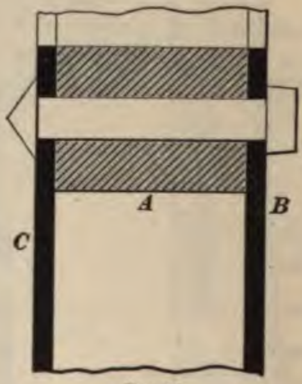


FIG. 24

the bottom of the water leg, the furnace plate is flanged and riveted to the shell, as shown.

CALKING

9. An upsetting process applied to a riveted joint, in order to make it steam-tight, is known as **calking**. The operation is shown in Fig. 25. A round-nose calking tool is

driven against the beveled edge of the upper plate, forcing the metal into close contact with the lower plate, thus effectually closing the seam. A tool with a sharp edge should never be used, as it is liable to score the under plate, and thus lead to grooving. Although the calking of boiler seams was formerly done entirely by hand, it is now largely performed by the pneumatic calking hammer.

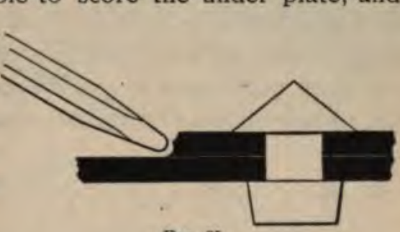


FIG. 25

When the edges of the plates have not previously been planed, it is necessary to chip them before calking. Formerly, this was done by hand; it is now performed by the pneumatic chipping hammer.

HEADS

FLAT HEADS

10. The **heads** of Scotch boilers are formed of one, two, or three sheets, according to the diameter of the boiler. The plates having been sheared and planed to the proper sizes and shapes, their curved edges are flanged over at right angles, as shown at *a*, *a*, Fig. 26 (*a*). After heating the plate in a furnace or forge at the edge to be flanged, the flange is turned over by either a hydraulic flanging press, a steam hammer, or, up to a certain thickness, by hand with large wooden mauls, the plate meanwhile resting on a cast-iron mold block of the required shape of the flange. The flanges are made of sufficient width to provide space for a single row of rivets in heads for boilers of small diameters, and for a double row of rivets in heads intended for boilers of large diameters. When the head consists of more than one sheet, the several sheets are joined by riveted lap joints, or seams, as shown at *b*, *b*, Fig. 26 (*b*). After the sheets, or plates, have been shaped, flanged, and fitted, the rivet holes for the horizontal seams are marked off and enough of them are drilled

to permit the sheets to be firmly bolted together, care being taken to drill the holes in the two adjoining sheets so that they will come fair with each other. After the sheets are bolted together, the rest of the rivet holes are drilled, either by a power drill or by hand. Of course, in all well-equipped boiler shops, all work that can be properly done by machinery is so performed in preference to hand work, and since the introduction of portable pneumatic tools much of the

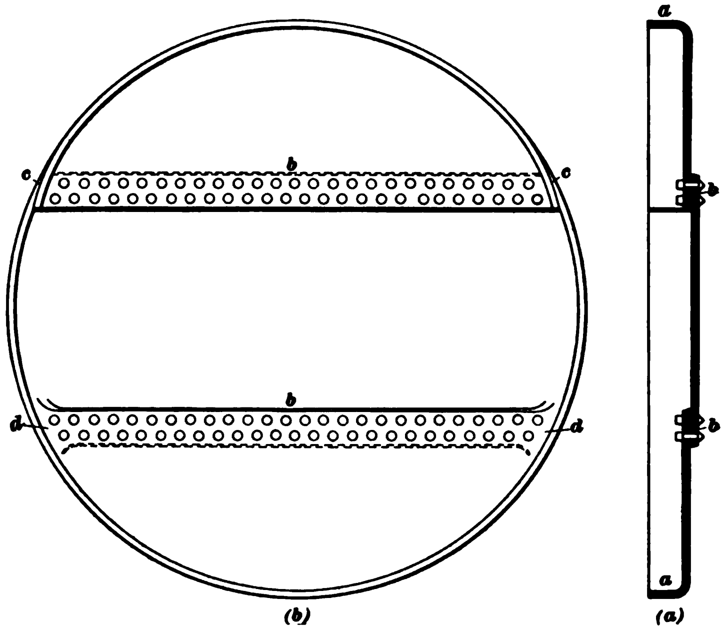


Fig. 26

work on boilers that was formerly done by hand is now accomplished mechanically by such tools.

At the ends of the lap joints in the head where the two flanges overlap, the flange of the under plate is forged tapering in order to make the joint at those points steam- and water-tight, as shown at *c*, *c*, Fig. 26 (*b*). In some cases, the overlapping flanges at the ends of the horizontal seams are welded together, as shown at *d*, *d*, Fig. 26 (*b*). This method insures a tight joint and a good fit of the head in the

shell, but it is objectionable owing to the uncertainty of making a reliable weld.

11. An elevation and a sectional view of the front head of either a single-ended or a double-ended boiler are illustrated in Fig. 27. The course of procedure of flanging, drilling,

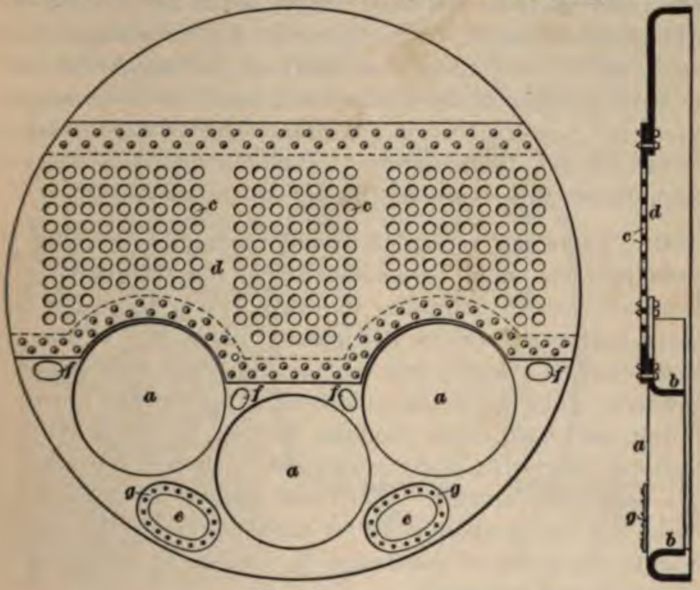


FIG. 27

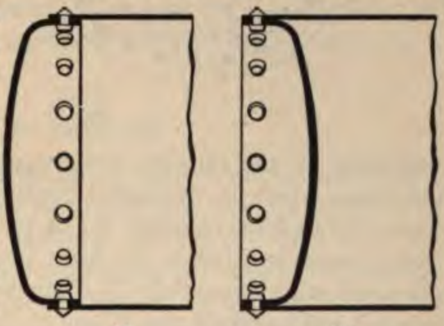
riveting, etc. in constructing this head is similar to that employed in the construction of the rear head, as just described. In addition thereto, the openings *a*, *a* for the furnace flues are cut out and flanged, as shown at *b*, *b*. The tube holes *c*, *c* are also drilled in the tube-sheet *d*, and the manholes *e*, *e* and handholes *f*, *f* are cut out. The manholes are reenforced by riveting a ring of boiler plate around them, as shown at *g*, *g*. All of these operations are performed before the plates are riveted together. The plates are also annealed before being riveted up.

12. Forming the front heads of a Scotch boiler is one of the most difficult operations attending its construction. This is owing to the deformation of the plates that takes

BUMPED OR DISHED HEADS

15. Cylindrical flue boilers of small diameters, such as are in service on the Red River of the North, North America, and rivers whose waters flow into the Gulf of Mexico, and steam, water, and mud-drums of all boilers, are usually fitted with **bumped or dished heads.** They may be either convexed or concaved; the radius of the curve to which they are bumped is usually equal to the diameter of the boiler or drum for which they are made.

Bumped heads are illustrated in Fig. 28, the one shown in Fig. 28 (*a*) being a convexed head and that shown in Fig. 28 (*b*) being a concaved head. They are flanged and riveted in the shell, as shown in the figure. These heads require no bracing, because they are bumped to the shape they would naturally assume under pressure; hence, they are self-supporting.



(*a*) (*b*)

FIG. 28

Bumped heads may have a manhole opening flanged inwardly when such flange has a sufficient depth and thickness to furnish as many cubic inches of material as was removed from the head to form the opening.

OPENINGS

MANHOLES

16. For the purposes of allowing the inside of the boiler to be inspected, cleaned, and repaired, holes closed by suitable covers are cut into the heads and shell. When they are of sufficient size to admit a man, they are called *manholes;* otherwise, *handholes.*

A common form of construction of a **manhole** and its cover is shown in Fig. 29. An elliptic hole is cut into the head or the shell of the boiler. A wrought-iron or steel ring *R*, called a *reenforcing ring*, is riveted to the plate *P*,

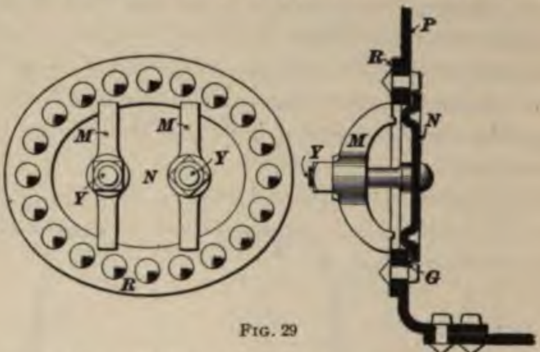


FIG. 29

generally on the outside, for the purpose of strengthening the plate, which is weakened considerably by the cutting of such a large hole through it. A cover *N* made of wrought iron or steel is fitted to the hole, inside of the boiler, and is provided with two studs *Y*, *Y* riveted to it. This cover is flanged and overlaps the edges of the plate about 1 inch or more all around its perimeter. A yoke *M* is slipped over each stud, its two extremities resting on the reenforcing ring. A ring *G*, or *gasket*, as it is commonly called, made of sheet rubber or any other pliable waterproof material, is placed between the plate and the cover and serves to make a water-tight joint.

17. Of late years, it has become quite generally the practice to flange the head inwards and face its edge, thus doing away with the necessity for the reenforcing ring. When the manhole is in the shell, in the best modern practice, a flanged reenforcing ring is riveted to the inside of the shell, as shown in Fig. 30. In this design, the edges of the ring and cover are faced and carefully fitted to each

FIG. 30

other, thus making a metallic joint. Owing to the practical difficulty of making such a joint perfectly water-tight, most engineers prefer to place a gasket, either fibrous or metallic, between the cover and its seat, even when both are faced and fitted to each other. The rules and regulations of the United States Board of Supervising Inspectors provide that all manholes for the shells of boilers over 40 inches in diameter shall have an opening not less than 11 inches by 15 inches in the clear, except that boilers with 40 inches diameter of shell or under shall have a clear opening in the manholes of not less than 9 inches by 15 inches. A manhole opening in the front head of externally fired boilers and under the flues must measure not less than 8 inches by 12 inches in the clear.

HANDHOLES

18. Handholes are placed in boilers whose construction does not permit the entrance of a man, as, for example, in vertical boilers. They are also placed in other boilers in convenient positions; thus, in boilers of the locomotive type, they are usually placed in the corners of the water legs, and in Scotch boilers they are placed above the crowns of the furnace flues. The handhole is a convenient place to rake out sediment and scale and to admit a hose for the purpose of washing out the boiler. The handhole and its cover are constructed very much like a manhole and cover; the handhole, being smaller, requires but one yoke and bolt to secure the cover.

Manholes and handholes are made elliptic to allow the cover to be passed through the hole. The smallest diameter of the cover is somewhat less than the largest diameter of the manhole, and thus allows the cover to pass freely through the manhole. It is then turned one-quarter around inside the boiler, the gasket placed on the flange, and put in position.

19. When using sheet rubber or other fibrous gaskets, it is advisable to give them a good coating of plumbago on both sides. This will prevent their sticking to the cover and

seat, thus allowing them to be readily removed. It is rarely advisable to use the same gasket again when replacing the cover; it will usually have become carbonized by the heat and thus be too hard to make a tight joint, no matter how hard the nuts are screwed up. When the cover has been replaced with a new fibrous gasket, it is well to examine it again after steam has been raised, and tighten the nuts once more. A plentiful supply of graphite (plumbago) smeared on the threads of the bolts before the cover is replaced will allow the nut to be readily removed at the next examination.

When a manhole or handhole gasket blows out, as will happen if the work of replacing the cover has been carelessly done or the gasket has been cut too large, about the only thing that can be done is to haul the fire, blow out the boiler after the steam has gone down, and make the joint over again.

Before taking off a manhole or handhole cover, immediately after the boiler has cooled down, it is advisable to raise the safety valve or open a valve or gauge-cock so as to break the vacuum that may have been formed by the condensation of the steam remaining in the boiler. If this precaution is neglected, it may result in serious injury. While it cannot be truthfully stated that a vacuum will always form, instances are on record where this has happened and the cover forced inwards by the external air pressure.

MISCELLANEOUS OPENINGS

20. Openings, other than manholes and handholes, are cut into the shells, heads, steam drums, and other parts of boilers to provide passages for steam or water to flow into or out of the boiler. Pipes and suitable valves are attached at the openings, either by screw threads or by flanged joints. Valves and cocks up to $1\frac{1}{2}$ inches in diameter can be secured by tapping the hole in the boiler with a pipe thread and screwing in these fittings; larger valves and cocks must be attached with flanges. Feedpipes and steam pipes up to 1 inch in diameter, if attached to a plate less than $\frac{1}{2}$ inch thick,

must be screwed into a bushing threaded inside and outside and screwed into the plate up to a shoulder provided on the bushing, which should preferably be secured with a jam nut placed on the water or steam side of the plate. Feedpipes and steam pipes up to 2 inches in diameter can be screwed directly into plates if they are $\frac{1}{2}$ inch or more in thickness. All pipes over 3 inches in diameter must be attached to the boiler or its parts by flanges. All holes over 6 inches in diameter cut into a boiler must be reenforced with a reenforcing ring, except when such holes are cut into a flat surface, in which case the plate may be flanged inwards to a depth of not less than $1\frac{1}{2}$ inches and the reenforcing ring dispensed with. On boilers carrying a steam pressure of not over 75 pounds per square inch, a flanged cast-iron stop-valve placed over an opening more than 6 inches in diameter may be used as a reenforcement. Openings serving as a connection between the shell of a boiler and a mud-drum must not exceed 9 inches in diameter.

APPURTENANCES

DOMES AND STEAM DRUMS

21. Domes are placed on cylindrical boilers for the purpose of increasing the steam space, and also for the purpose of drying the steam, the supposition being that the steam will be dried on account of its being farther removed from the water. The hole cut into the shell to give communication between the boiler and the dome should be made only large enough to allow a man to pass through, since a large hole materially weakens the shell. The edge of the plate around the hole should be reenforced by a wrought-iron ring riveted to it. The flat top of the dome must be stayed by diagonal braces. Steam domes usually have a diameter equal to one-half the diameter of the boiler, and a height equal to about nine-sixteenths the diameter of the boiler.

22. Shell boilers are often fitted with a **steam drum** instead of a dome. The steam drum is simply a cylindrical

vessel connected to the shell. When several boilers are set so as to form a battery, they are often connected to one drum common to all boilers. When each boiler has its own furnace, there should be a stop-valve between each boiler and the drum to allow the boiler to be taken out of service when required. When the boilers in battery have one furnace common to all of them, no stop-valve should ever be placed in the pipe connections between each boiler and the drum. Where boilers are in battery with separate furnaces, each boiler must have its own safety valve, which should always be so fitted that it cannot be cut off from the boiler under any circumstances. Scotch boilers are rarely fitted with steam drums; they are frequently used, however, in connection with the boilers installed on Western-river steamboats. Nearly all designs of water-tube boilers require a combined steam and water drum.

23. Some boilermakers, when fitting a longitudinal steam drum to a shell boiler, will attach it by two nozzles. Many engineers object to this method, since with an unequal expansion of the boiler and drum, which is quite likely to occur, the joints of the nozzles will become leaky, owing to the strains to which they are subjected. It is now the rule, in good work, to use one nozzle only. When the steam drum is used for a single boiler, its diameter may be made equal to one-half the diameter of the boiler, and its length equal to the diameter of the boiler. Where one steam drum is common to several boilers, its diameter is usually made equal to half the diameter of one of the boilers, and its length equal to the horizontal outside-to-outside measurement over the several boiler shells.

The strength of steam drums may be determined by the rules governing the strength of boiler shells. They require just as rigid inspection as the boiler itself.

MUD-DRUMS

24. Mud-drums are occasionally attached to boilers for the purpose of providing a quiet place for the collection of mud and sediment in mechanical suspension in the feedwater, which is introduced into the mud-drum. In shell boilers the mud-drum is located underneath the boiler and at the rear end, being connected to the boiler by a suitable nozzle. When several boilers are set in battery, they are sometimes connected to a common mud-drum. This practice is permissible when the whole battery is used at once. When so fitted, none of the boilers can be temporarily taken out of service unless each nozzle is provided with a stop-valve. Owing to the difficulty of protecting the valve from the heat of the fire, this is rarely if ever done. This consideration limits the use of a common mud-drum to cases where all the boilers are worked together. When a mud-drum is fitted, the blow-off should be attached to it and the sediment collected in the drum should frequently be blown out.

Mud-drums for shell boilers are not used to any extent outside of Western-river steamboats and vessels engaged in similar service. Practically all marine water-tube boilers have one or more mud-drums or the equivalent thereof, their design generally requiring this.

STAYING

PURPOSE AND CLASSIFICATION

25. The surfaces of boiler shells are, in general, either cylindrical, hemispherical, or flat. A cylinder or sphere subjected to an internal steam pressure is *self-supporting;* that is, the steam pressure tends to maintain the cylindrical or spherical form of the vessel, and hinders distortion instead of producing it. If, on the contrary, the vessel is composed of flat surfaces, the steam pressure tends to distort it and give it an approximately spherical form. Hence, flat surfaces are not self-supporting, and must be braced or stayed.

The flat surfaces commonly found in boiler construction are the flat heads of Scotch boilers and the sides, back, and top of the combustion chambers; in firebox boilers, the sides, back, and top of the combustion chambers, the crown of the furnace, the water legs, and the flat heads.

The appliances used for bracing steam boilers may be divided into *direct stays*, *diagonal stays*, and *girder stays*.

A **direct stay** may be defined as one in tension, in which the load is applied directly in line with the axis of the stay.

A **diagonal stay** is a tension member in which the load acts at an inclination to the stay; in other words, it is a stay that is not placed at right angles to the surfaces it supports.

A **girder stay** is a stay in the form of a girder, and is subjected to bending stresses produced by the load.

DIRECT STAYS

26. Screw Stays.—The most common form of a **screw stay** used in firebox boilers is shown in Fig. 31. The stay

FIG. 31

is threaded the entire length. It is screwed into place and the ends are headed over by hammering. A much better form is shown in Fig. 32. The thread is turned off in the center,

FIG. 32

which increases the durability of the bolt, for the reason that a smooth surface is not so readily attacked by corrosion as a threaded surface.

27. Staybolts.—Fig. 33 shows a **staybolt** of the construction usually met with in Scotch boilers, used for staying the sides and back of the combustion chambers. It consists

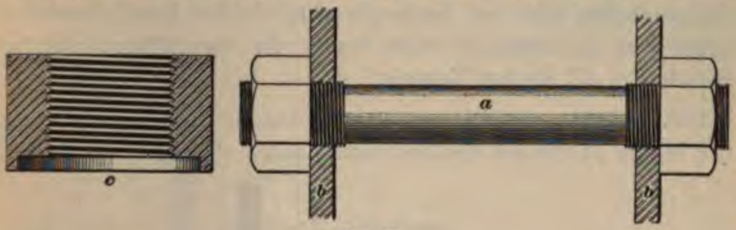


FIG. 33

of a wrought-iron or steel bolt *a*, screwed into the two plates *b*, *b*, and secured by a nut at each end. An enlarged section of the nut is shown at *c*, Fig. 33. The face of the nut is recessed, the recess being filled with red-lead putty mixed with iron filings. The putty serves to make a steam-tight and water-tight joint.

All screw stays and staybolts for boilers using fresh water that are constructed in the United States after July 1, 1899, must be drilled on both ends with a central hole, as *a*, Fig. 32, having a diameter of not less than $\frac{1}{8}$ inch and a depth sufficient to extend at least $\frac{1}{2}$ inch beyond the inside surface of the sheet. Should a stay break, water will issue from this hole, thus giving notice of the break.

28. A **socket staybolt** is shown in Fig. 34. The

FIG. 34

socket consists of a tube expanded into the sheets *b* and *c*. A bolt *d* fitting closely the inside of the tube *a* is passed through and secured by a nut. The ends of the socket project from the sheets and are beaded over. Sometimes the

staybolt shown in Fig. 31 is protected by a socket. If salt water is used for the generation of steam, all screw staybolts must be provided with sockets to protect them from the corrosive effect of the sea-water. Water from a surface condenser is deemed fresh water by the United States Inspectors of Steam Vessels.

29. Stayrods.—A rod that is chiefly used for staying the heads of Scotch boilers, and passes through from head

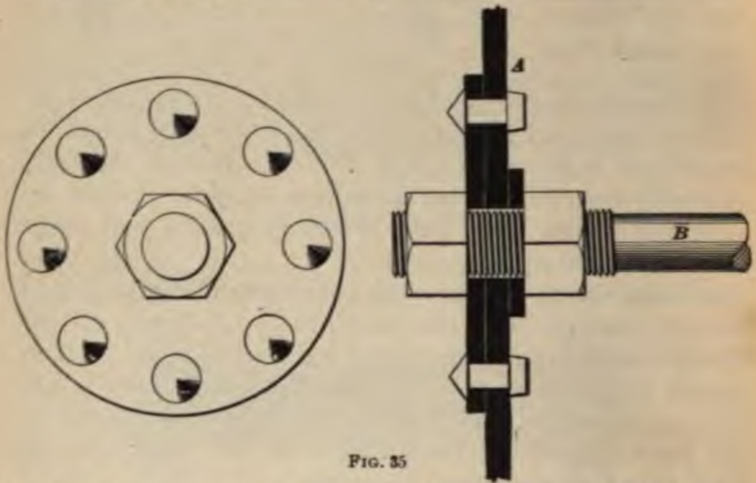


FIG. 35

to head, is known as a **stayrod**. Examples of stayrods are shown in Figs. 35, 36, 37, 38, and 39. The end of the stayrod *B*, Fig. 35, is enlarged and threaded and passes through the plate *A*. Two nuts and washers are provided. The larger washer is on the outside and is riveted to the plate; it thus serves to distribute the supporting effect of the rod.

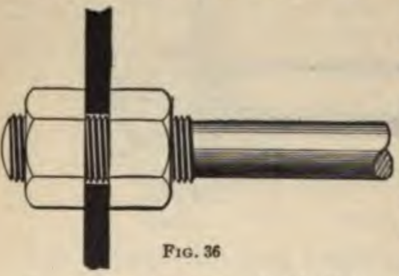


FIG. 36

Sometimes, instead of the washer, a stiffening plate is used, covering the whole area to be braced, and placed either inside or outside of the boiler. By means of the nuts,

the tension of the stayrod may be adjusted, the nuts on the inside serving to lock the rod in position.

Sometimes stayrods without washers, as shown in Fig. 36, are employed. The nuts in this case are recessed, the same as those illustrated in Fig. 33. Occasionally, two small washers are used, as shown at *a* and *b*, Fig. 37. A some-

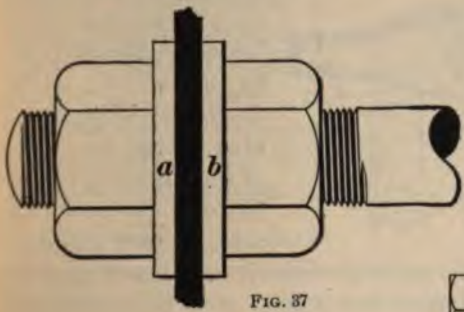
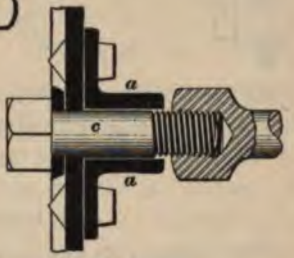


FIG. 37



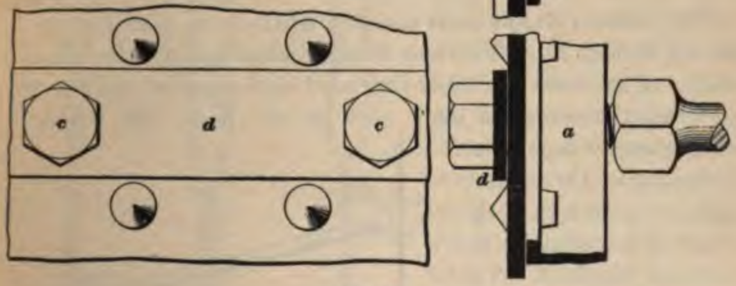


FIG. 38

what different form of a stayrod is shown in Fig. 38. Two angle irons *a*, *a* are riveted to the plate. The end of the stayrod is enlarged and made square in cross-section. It is tapped to receive a bolt *c* passing through the plate and between the angle irons. A leaf *d* is placed on the outside and helps to support the plate.

The diameters of through stayrods vary from $1\frac{3}{4}$ to $2\frac{3}{4}$ inches, according to the steam pressure.

Another method of connecting the ends of stayrods to the plate is shown in Fig. 39. Here the end of the stayrod A, instead of being threaded, is forked to receive the connection B B, which is riveted to the sheet to be stayed, and joined to the stayrod by a bolt C. The combined effective area of the two legs of the connection B B should exceed the area of the stayrod. This connection is often used to support the lower part of the rear tube-sheets of Scotch boilers.

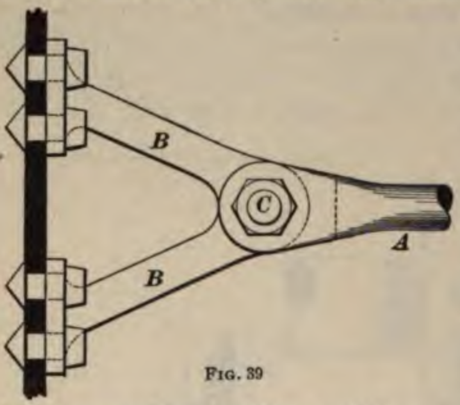


FIG. 39

DIAGONAL STAYS

30. Palm Stays and Gusset Stays.—A **palm stay** is shown in Fig. 40. The end B is flattened and riveted to the shell. The other end A is threaded and supplied with a nut and taper washer on each side of the head, the washers having such a taper that, when one of the faces is against the head, the other is parallel to the face of the nut. The hole through which the stay passes is not threaded, but is made sufficiently large to allow the stay to pass through. In some instances, the end A is bent, so as to pass through the plate at a right angle to it. This is done to obviate the need of taper washers.

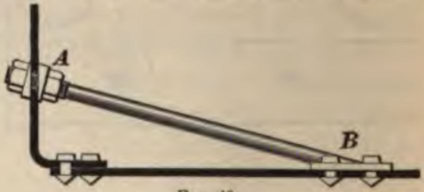


FIG. 40

Palm stays, or **diagonal stays**, as they are sometimes called, are used in locations prohibiting the use of a through stayrod. In Scotch boilers, they are usually found supporting

the lower part of the heads between the furnaces. The combined area of the rivets attaching the stay to the shell should at least equal the area of the rod. The angle that a diagonal stay makes with the shell should not exceed 30°, and should be as much smaller as possible.

31. In Fig. 41 is shown a **gusset stay.** This stay consists of a wrought-iron or steel plate *A*, secured to the head and shell by either angle or **T** irons *B*, *B*. Gusset stays are sometimes used for the same purpose as palm stays.

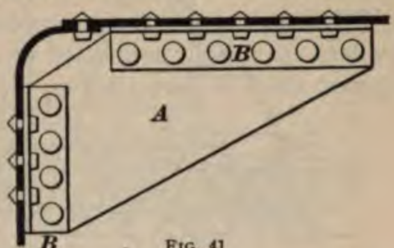


Fig. 41

32. Crowfoot Braces.—The flat heads of boilers are sometimes supported by **crowfoot braces,** which are securely riveted to the head and the shell. The crowfoot brace is shown in Fig. 42 (*a*). In this style of brace, the crowfoot, or part that is riveted to the head, is formed by welding flat bars to a cylindrical stem. The strap end, or part that is riveted to the shell, is also welded to the stem. An improved form of crowfoot brace is the **McGregor brace,** shown in Fig. 42 (*b*). This brace is formed by a piece of sheet steel, bent in one heat as shown. Being weldless, it may naturally be assumed, and the assumption has been borne out by experiments, that, for equal cross-sectional areas, it will bear a much greater strain than the welded crowfoot brace. It will be observed that the crowfoot of the McGregor brace is formed by

Fig. 42

splitting the sheet and bending it at a right angle. In the **Huston** improved crowfoot brace, shown in Fig. 42 (*c*), the crowfoot is formed by flanging the plate of which the brace is formed, thus giving probably the strongest form of crowfoot that can be devised.

GIRDER STAYS

33. The tops of the combustion chambers of Scotch boilers are usually supported by **girder stays,** the construction of which is shown in Fig. 43. Two girders A, A' are held

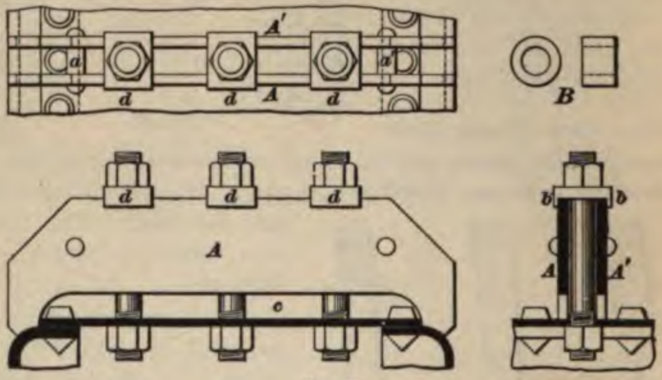


FIG. 43

apart by two distance pieces a, a'. Staybolts, similar to the one shown in Fig. 33, are supported by the girders. To prevent spreading of the girders, washers d, d provided with lugs b, b are used. To give access to the plate and to prevent local overheating of the plate, which would likely occur if the girders touched the whole length of the plate, a space c, generally about 2 inches in depth, is left between the girder and the plate. To prevent buckling of the plate by setting the staybolt too tight, a thimble or socket B is sometimes placed over the staybolt between the girders and the plate.

34. The upper plates or crown sheets of the furnaces of internally fired boilers of the locomotive type are supported by girder stays, or **crown bars,** as they are usually called when applied to a firebox boiler. These are sometimes

further supported by **sling stays**, which consist of brace rods running from the crown bars to an angle bar riveted to the shell of the boiler above the crown bars.

Referring to Fig. 44, A is the crown bar, B, B are the side

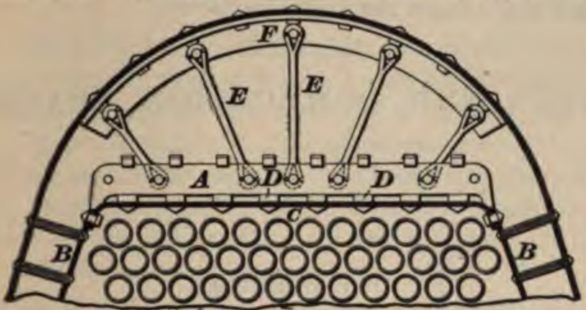


FIG. 44

sheets of the firebox, C is the crown sheet, D, D are the pins that secure the crown bar to the crown sheet, E, E are the sling stays, and F is the angle bar. The latter is bent to the shape of the boiler shell and securely riveted to it, as shown in the

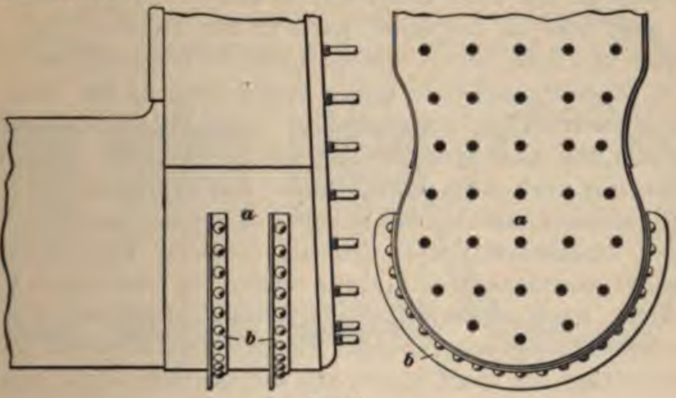


FIG. 45

illustration. The lower ends of the sling stays have single eyes forged in them that are secured between the two members of the crown bar by bolts or split pins. The upper ends of the sling stays are forked and have two eyes that straddle the flange of the angle bar and are secured to it by bolts or pins.

35. The lower part of the combustion chambers of Scotch boilers is often strengthened by several hoops made of angle iron and riveted to the plate. This arrangement is shown in Fig. 45, in which *a* is the lower part of the combustion chamber and *b*, *b* are the angle-bar hoops.

FIRE AND COMBUSTION SPACES

FURNACES

INTRODUCTION

36. Furnaces of marine boilers may be divided into two distinct classes, namely, *internal furnaces* and *external furnaces*. The **internal furnaces** of cylindrical or shell boilers of the Scotch, firebox, locomotive, and vertical types are parts of the apparatus, being built into the boilers during their construction, and of similar material to that used in the other parts of the boilers. The furnaces of these boilers are portions of the pressure parts and are surrounded, or partly surrounded, by the water in the boiler. They are therefore built strong enough to sustain the same pressure to which the boiler shell and other pressure parts of the boiler are subjected.

Cylindrical marine boilers, such as are used on Western-river steamboats, have external furnaces, and they are usually constructed of firebrick under the front end of the boiler. Brick furnaces are not suitable, however, for the boilers of sea-going vessels. The brick walls are too heavy and occupy too much space, and the working of the ship in a seaway would cause them to crack and eventually fall.

The furnaces of marine water-tube boilers like the Babcock & Wilcox, Almy, Roberts, and other steam generators of this type may be properly called internal furnaces, as they are surrounded on their sides by water tubes that form a part of the heating surface, and are in communication with the

steam space. The generating tubes of water-tube boilers are enclosed in a casing of sheet iron or steel, lined with some refractory substance, such as asbestos, magnesia, etc.

FURNACE FLUES

37. The furnaces of Scotch boilers are cylindrical in form, and are known as **furnace flues.** The longitudinal seams of furnace flues may be either welded or riveted, the latter method being now obsolete. When furnace flues with riveted longitudinal seams are used, they are fitted into the boiler so that the seams are below the grate bars, in order that the fierce heat of the fire may not injure the seams. Furnace flues are either *plain* or *corrugated.*

38. Plain furnace flues may be made in different ways, as shown in Fig. 46. The flue shown at *A*, Fig. 46, is made in sections of not more than 8 feet in length. Each section is flanged to a depth of not less than $2\frac{1}{2}$ inches, and the sections are riveted together with a wrought-iron ring (shown at *a*) between the flanges. The thickness of the ring must not be less than $\frac{1}{2}$ inch nor its width less than $2\frac{1}{2}$ inches.

A different construction is shown at *B*, Fig. 46. Angle-

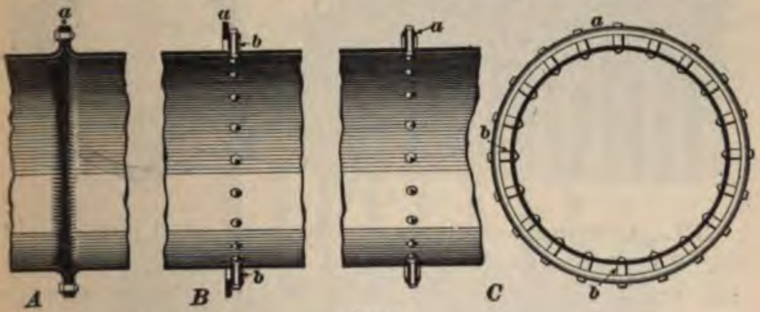


FIG. 46

iron rings, as *a*, are employed, serving to stiffen the flue. The thickness of the material of the ring must not be less than double that of the flues, and the depth must not be less than $2\frac{1}{2}$ inches. The rings are held in position by rivets passing through wrought-iron thimbles *b*, *b*, placed between

the inner surface of the strengthening ring and the outer surface of the flue. The length of the thimbles must not be more than 2 inches, nor the diameter of the rivets less than $1\frac{1}{2}$ times the thickness of material in the flue. The pitch of the rivets, measured at the outer surface of the flue, is not to be more than 6 inches.

At *C*, Fig. 46, the strengthening ring shown at *a* is made of half-round iron. The proper area of the ring may be found by multiplying the thickness of the flue in decimals of an inch by the constant 9.6; the product will be the area, in square inches. The ring is held in position by rivets passing through wrought-iron thimbles *b* not more than 2 inches in height. If rivets $\frac{7}{8}$ inch diameter and over are used, the pitch must not exceed 8 inches; for rivets $\frac{3}{4}$ inch diameter, 6 inches; or for rivets $\frac{5}{8}$ inch diameter, 4 inches; the pitch to be measured at the outer surface of the flue. No rivets smaller than $\frac{5}{8}$ inch are to be used for securing strengthening rings.

The distance from center to center of flanges or strengthening rings is to be taken as the length of the flue in computing the working pressure allowable.

FIG. 47

39. Corrugated furnace flues are used extensively for Scotch boilers, the corrugation serving to strengthen the flue considerably, and they also permit the flue to expand and contract freely in the direction of its length without subjecting the combustion chamber and the front head of the boiler to undue strain, as is the case with the plain furnace. The corrugations open and close by expansion and contraction like the bellows of an accordion, but, of course, to a much less degree. The longitudinal seams of corrugated

furnaces are invariably welded. In Fig. 47, a common construction of such a flue is shown. The end *A* is cylindrical and is attached, by rivets, to the head of the boiler, which is flanged either inwardly or outwardly to fit closely over the

FIG. 48

cylindrical part of the furnace flue. The other end of the flue is flanged in the manner shown; the back tube sheet is riveted to the flange *C*, and the plates forming the sides of the combustion chamber are riveted to the flanges *D*, *D*. The distance from center to center of corrugations is 8 inches; the plain part at the ends must be not more than 9 inches, and the thickness not less than $\frac{5}{16}$ inch. The radius of the outer corrugation must be not more than half that of the reverse or suspension curve.

40. A **Purves ribbed furnace flue** is shown in Fig. 48. The height of the ribs is made $1\frac{3}{8}$ inches, the distance from

FIG. 49

center to center of ribs 9 inches. The thickness of the flue must not be less than $\frac{7}{16}$ inch, and the length of the plain part at the ends must not exceed 9 inches.

Both corrugated and ribbed flues may be used for the uptakes of boilers having a wet uptake.

41. A **Morison suspension furnace flue** is shown in Fig. 49. This flue somewhat resembles the corrugated flue shown in Fig. 47. The outer corrugations, as *a*, *a*, Fig. 49, are made to a small radius, and are joined by a curve of large radius.

FURNACE FITTINGS

42. General Arrangement.—The **furnace fittings** of a Scotch boiler are shown in Fig. 50. A furnace front *A*, made of cast iron, is fitted to the front of the furnace. It is

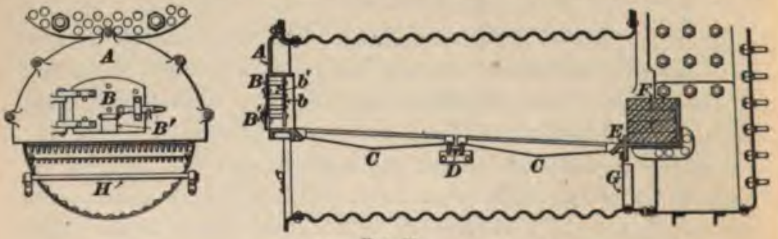


FIG. 50

held in position by studs screwed into the head of the boiler. A furnace door *B* is fitted to the furnace front. It is generally made of the shape shown, and is about 18 inches by 15 inches in size. A baffle plate *b* is attached to the door by means of long bolts, distance pieces *b'* made of iron pipe serving to keep the plate in place. This baffle plate increases the durability of the furnace door and prevents, to some extent, the radiation of heat, as it absorbs most of the radiant heat of the burning fuel, and thus prevents its coming in contact with the door. The furnace door is often provided with a small hinged door *B'* to allow the slice bar to be pushed into the furnace and the fire sliced without opening the furnace door. The inrush of cold air that always accompanies the opening of the door proper is thus avoided. Means for admitting air above the grate are often provided. The furnace door may be perforated and supplied with a suitable arrangement by which the admission of air may be regulated.

Baffle plate *b* may also be perforated to aid in distributing the air. The lower end of the furnace front forms what is known as the *dead plate*, extending clear across the furnace.

43. Grate.—The **grate**, which is generally made in two sections, as *C, C*, Fig. 50, is composed of cast-iron grate bars supported by a bearing bar *D* in the center of the furnace and by the dead plate and the bridge bar *E*.

The grate bars are generally made from 3 feet to 3 feet 6 inches in length, a greater length being awkward to handle. In order to facilitate the access of air, the fall of ashes, and the cleaning of the fire from below, they are made somewhat thinner at the bottom. They are provided with distance pieces cast with the bar at both ends, and sometimes also in the center. These keep the bars apart.

Grate bars for marine boilers are often cast with two ribs, and when so made are called *double bars*. Single bars, with one rib, are used only to fill out a row of bars when there is not space enough to insert a double bar. A double grate bar is illustrated in Fig. 51. The grooves *a, a* are cast along the top edges of the ribs. These grooves soon become

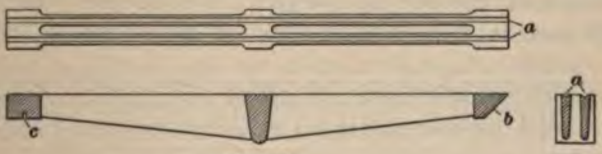


FIG. 51

filled with ashes or clinkers, which protect the top of the bar from being burnt. In order to allow grate bars to expand freely in the direction of their lengths, one end of each bar, where it rests on the dead plate or bridge-wall plate, is made slanting at an angle of about 45°, as shown at *b*, Fig. 51. The slanting end of the bar is in contact with a similar slant in the dead plate or bridge-wall plate. This permits the slanting ends of the bars to slide up when expanded. If both ends of the bar were made square, there would be no room for the bar to expand and it would soon become bent out of shape. Even if spaces were left between

the ends of square bars and the dead plate or bridge-wall plate, they would soon become filled with ashes and their object thereby defeated. End motion of the bar is prevented by notching the end of the grate bar that rests on the center bearing bar, as shown at *c*, Fig. 51. The grate is usually placed lower at the back of the furnace, the inclination being about $\frac{3}{4}$ inch for each foot of length of the grate. This permits free access of air to the back of the grate.

44. The width of the air space and hence the thickness of the grate bar depend largely on the character of the fuel burned. For the larger sizes of anthracite and bituminous coals, the air space may be from $\frac{5}{8}$ to $\frac{3}{4}$ inch wide, and the grate bar may have the same width. For pea and nut coal, the air space may be from $\frac{3}{8}$ to $\frac{1}{2}$ inch, and for finely divided fuel, like buckwheat coal, rice coal, birdseye coal, culm, and slack, air spaces from $\frac{3}{16}$ to $\frac{3}{8}$ inch may be used. When these small air spaces are used, the grate, if made of bars like that shown in Fig. 51, must have the ribs so thin in proportion to their length that they will warp and twist, and a large number of the bars will soon break, especially when the rate of combustion is high. To overcome this objectionable feature, the grate bar shown in Fig. 52, and known as

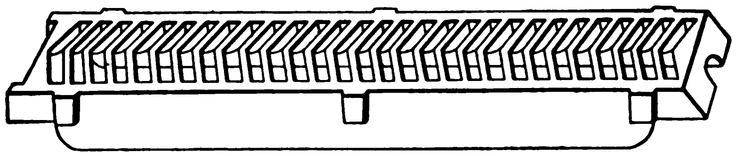


FIG. 52

the **herring-bone grate bar**, was designed, and where the small sizes of coal are used, it has almost entirely superseded the ordinary grate bar. Owing to the shape of the supports for the fire, they are free to expand and contract; being quite short and of small depth in comparison to the ordinary grate bar, there is very little danger of excessive warping of the supports. In consequence, they will usually far outlast a set of ordinary grate bars. Since there are only a few large bars for the grate, it is also easier to replace a broken bar.

Herring-bone grate bars can be obtained in a great variety of styles and with different widths of air spaces.

45. In general, a grate bar that is suited for the kind of fuel that is to be burned should be selected. Thus, if finely divided coal is to be burned, a grate bar having small air spaces and supports should be selected, since otherwise a large percentage of the fuel will fall into the ash-pit. On the other hand, for the large sizes of coal, select bars having large air spaces, using the largest air space when caking coals are to be burned. Some varieties of bituminous coal will *cake*, that is, fuse together to a considerable degree, and the ashes and clinkers formed will be of such size that a large part of them cannot pass through the air spaces unless these are ample; the grate thus becomes clogged, shutting off the air from the fire. This reduces the rate of combustion and evaporation.

46. Bridge and Ash-Pit.—Referring again to Fig. 50, the **bridge** F is built up of firebrick on a bearing bar of the cross-section shown. Its object is to retard the escape of the gases from the furnace, and thus promote a more perfect combustion. The distance from the top of the bridge to the top of the furnace varies from 13 inches to 16 inches. The best height is determined by actual trial, as it depends on the intensity of the draft, the thickness of the fire, the kind of coal used, and the quantity of air admitted. Generally speaking, the area over the bridge is equal to one-sixth the area of the grate. The cast-iron bar on which the bridge is built is attached to brackets made of angle iron riveted or secured by studs to the side plates of the combustion chamber. The lower half of the furnace flue, that is, the part below the grate that forms the **ash-pit,** is usually lined with sheet iron. Sometimes a thin coating of cement is applied. A baffle plate G is secured by tap bolts to the bridge bar E and prevents the free access of air to the combustion chamber. Sometimes this plate is perforated, as in some cases it is of advantage to admit air behind the bridge. If insufficient oxygen is mixed with the fuel, incomplete

combustion will result. But, by mixing oxygen with the products of combustion, after they have left the furnace proper, and while they are still in a red-hot state, complete combustion can be obtained. Sheet-iron ash-pit doors are usually provided for the furnaces. They are not attached to the furnace, but are kept in a convenient place until required, when they are merely placed in position, two hooks riveted to the door serving to keep it in place.

47. Lazy Bar.—A wrought-iron bar *H*, Fig. 50, extends across the furnace, about 12 inches below the dead plate. This bar serves as a rest for the fire-tools when cleaning the ash-pit, and a support for the pricker bar while pricking the fire from below, and is called a **lazy bar** by firemen. There is another form of lazy bar that is placed across the furnace-door opening as a support for the hoe and devil's claw while cleaning, banking, or hauling the fires. This lazy bar consists simply of a bar *a* of 1¼-inch round iron bent to the shape shown in Fig. 53, in which *b* represents the furnace-door opening, *c*, *c* are the hinge lugs attached to the furnace front, and *d* the catch of the door latch. It will be observed that one end of the bar rests on the upper hinge lug, while the other end is supported by the catch of the door latch, the furnace door, of course, being open while the bar is in position for use. The latch end of the bar may be flattened so that it will fit into the notch of the latch catch and at the same time form a shoulder that will prevent the bar from sliding endwise. This may also be accomplished by forming the hinge end of the bar into a hook that will grip the hinge lug. This simple arrangement enables the fireman to clean the fires more quickly and get the furnace doors shut sooner than without it, so that the rush of cold air into the furnace will be stopped earlier. It also lightens the labor of the fireman very

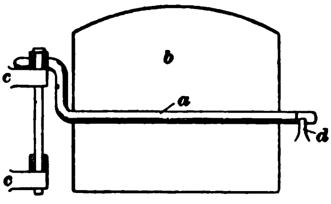


FIG. 53

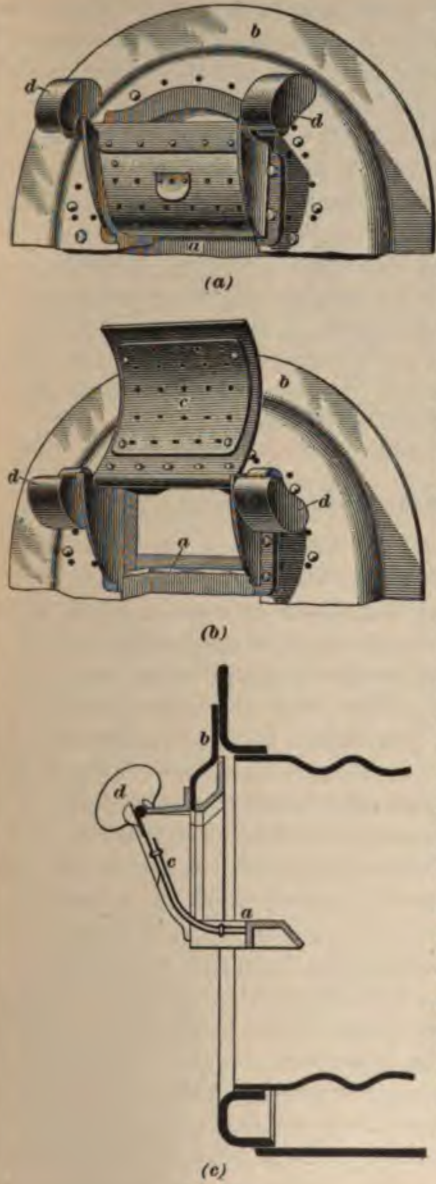


Fig. 54

considerably, especially if very heavy firing tools are used.

48. Furnace Door. The Morison furnace door illustrated in Fig. 54, is intended to overcome some of the defects that are inherent to the usual type of furnace doors in use on marine boilers. In the illustration, the door is shown closed at (*a*) and open at (*b*); a sectional view through the door and furnace front is given at (*c*). The primary object in constructing the door in this manner is to prevent the undue accumulation of fuel on the front end of the grate, which causes overheating and ultimate destruction of the furnace door and its attachments; in consequence of the freedom from obstruction in the front end of the furnace, very much better facilities are afforded for properly working the fire.

To accomplish this, a portion of the dead

plate immediately inside of the furnace door is cut away at *a* so as to leave a recess. The door is provided with an extension, which, when the door is closed, fills the recess in the dead plate. This extension, and also the vertical portion of the door, may be perforated for the admission of air into the furnace. The furnace front *b* is made of a plate of pressed steel worked to the shape indicated in the illustration; the furnace door is protected from the fire by the perforated liner *c*. This furnace door is arranged to open upwards, and is so counterbalanced by the weights *d*, *d* as to remain open while the furnace is being stoked. This is an advantageous feature in a marine boiler, as it does away with the necessity of catches or other devices for preventing the door from being closed by the motion of the ship.

COMBUSTION CHAMBERS

PURPOSE

49. The **combustion chamber** of a steam boiler is an enclosed space, usually located back of the bridge wall, providing a place for the unconsumed gases to be thoroughly mixed with air and thus effect their complete combustion. In some cases, a small quantity of air is admitted into the chamber from the ash-pit through small openings in the bridge wall or in the diaphragm below the bridge wall. In other cases, the air is admitted through small perforations in the furnace door. Sometimes, the excess of air that passes through the grates is depended on to produce the complete combustion of the gases in the combustion chamber; or, small openings may be made in the sides of the combustion chamber through which the air may enter. In all cases, however, provision should be made to regulate the quantity of air admitted, as more air is required to completely consume the gases under some conditions than under others. When bituminous coals are used, a large quantity of air will be required; while with the hard anthracite much less air will be needed. The lowest temperature at which

ignition of the gases can take place is about 1,800° F.; it is, therefore, evident that, if the gases are cooled below the point of ignition by too much air or otherwise, they will be carried to the smokestack unconsumed. It follows that the furnace must either be of sufficient height to provide a space in which the great volume of gas can burn before being cooled, or else there must be a combustion chamber adjacent to the furnace in which the gases can burn. No set rule can be made in regard to the right quantity of air to admit to the combustion chamber; it depends on the experience, skill, and attention of the firemen to obtain the best results.

CONSTRUCTION

50. The combustion chambers of internally fired marine boilers are constructed of similar material, and the plates are joined together by riveted seams in a similar manner to that employed in the construction of the shells and heads of boilers of that class. They are built into the boilers and are designed to sustain safely the steam pressures that are carried in the boilers to which they belong.

The shape of a combustion chamber depends on the form of the boiler of which it is a part, and as to whether or not it is internal or external. Internal combustion chambers are usually made circular at their lower ends to conform to the curve of the boiler shells, and concentric with them. The upper parts are usually rectangular with flat or arched tops. External combustion chambers are usually rectangular or nearly so.

51. A transverse section of the longitudinal section given in Fig. 55 (*a*), of two of the combustion chambers of a four-furnace, single-ended, Scotch boiler is shown in Fig. 55 (*b*). The sectional view given in Fig. 55 (*a*) is taken through the center of the combustion chamber *a*, Fig. 55 (*b*). By the method of construction shown, each furnace is provided with its own combustion chamber, which arrangement is preferable to having one combustion chamber common to two or more furnaces, though more expensive to build. The front sheet

of the combustion chamber, which is also the tube-sheet, is shown at *b*, and the rear sheet at *c*; *d* is the furnace flue. The tube-sheet and rear sheet are flanged inwardly, as shown at *e*, *e*, to which flanges the side and top sheets *f*, *f* are riveted. A circular opening is cut in the lower part of the tube-sheet to receive the rear end of the furnace flue, the

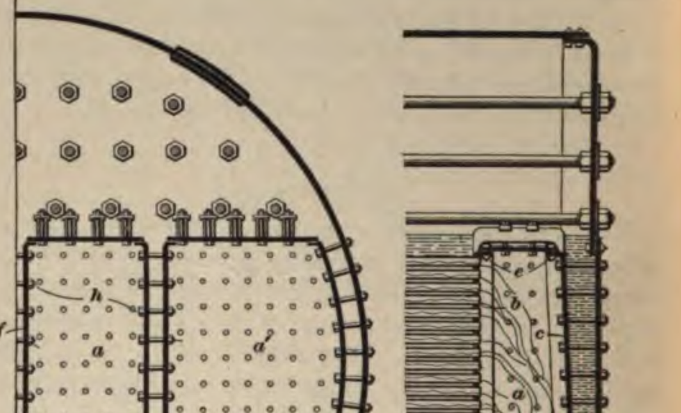


FIG. 55

two being firmly riveted together, as shown at *g*, *g*. Combustion chambers of Scotch boilers are secured to the shell and rear head of the boiler and to each other by staybolts, as *h*, *h*. The bridge wall *i*, which is constructed of firebrick, is built on the cast-iron bearing bar *j*. The brickwork extends across the floor of the combustion chamber and up the rear sheet of the same for some distance above

the top of the furnace flue, as shown at *k* and *l*, to protect the metal at those points from the intense heat of the flame, which otherwise would impinge directly against it, accomplishing its early destruction.

52. Combustion chambers are sometimes constructed with rounded or arched backs, as shown at *a*, Fig. 56. The

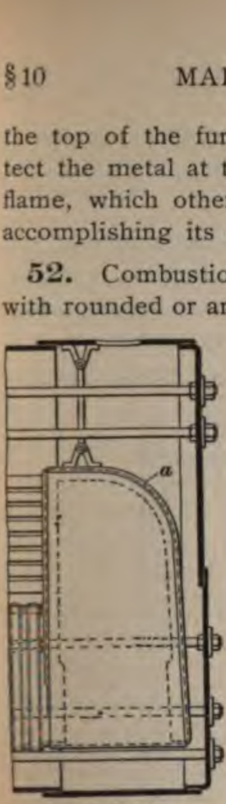


Fig. 56

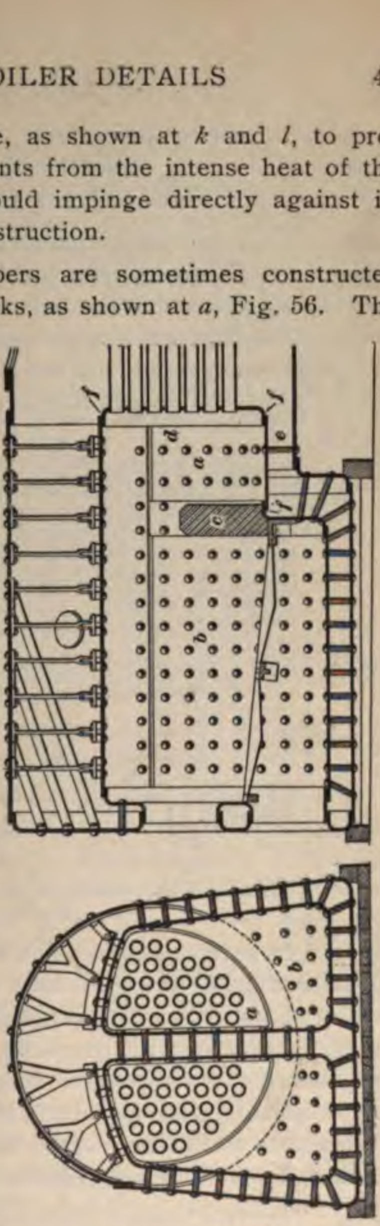


Fig. 57

purpose of this is to facilitate the flow of the gases of combustion into the tubes, the curved top of the combustion chamber acting as a deflector for the gases.

53. The combustion chambers of firebox boilers of the locomotive type are constructed as shown in Fig. 57. It will be observed that in this boiler the combustion chamber *a* is an extension of the furnace *b* and is separated from it by the bridge wall *c*; also, that the back sheet of the combustion

chamber is the front tube-sheet. The side sheets are secured to the shell of the boiler by the staybolts *e*, *e*, and to the tube-sheet and firebox by the flanges *f*, *f*.

54. The combustion chamber of a firebox boiler of the flue and return-tubular type is shown at *a*, Fig. 58. It is constructed and secured in the boiler in very much the same way as those just described, with this difference, however: it is located at the rear end of the flues *b*, *b*, at some distance

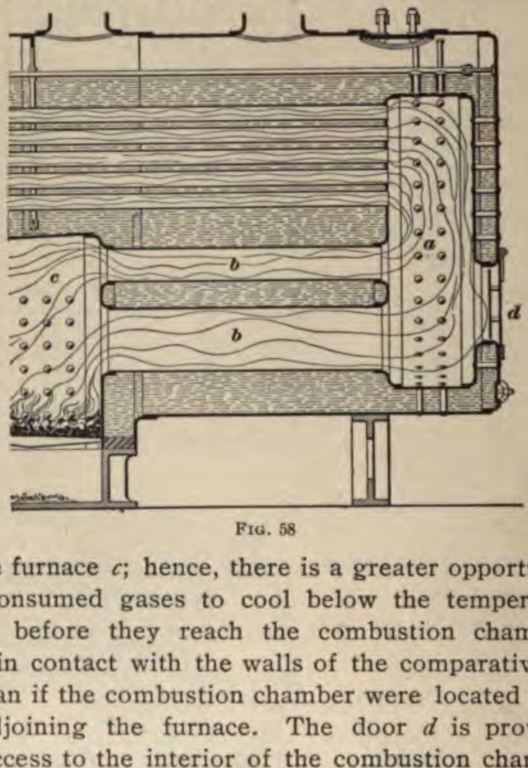


FIG. 58

from the furnace *c*; hence, there is a greater opportunity for the unconsumed gases to cool below the temperature of ignition before they reach the combustion chamber, by coming in contact with the walls of the comparatively cool flues, than if the combustion chamber were located immediately adjoining the furnace. The door *d* is provided to afford access to the interior of the combustion chamber for cleaning, inspection, and repairs.

55. The combustion chamber of a return-flue boiler is illustrated at *a*, Fig. 59. This is an externally fired boiler in which the construction of the combustion chamber differs radically from those of internally fired boilers. In this case,

it is formed of firebricks. This method of construction has its peculiar advantage. The combustion chamber being located at a considerable distance from the furnace *b*, the unconsumed gases would be cooled below the temperature of ignition if it were not for the fact that the bricks of which the combustion chamber and the smoke flue *c* are constructed become highly heated and the heat given off from them assists in keeping the temperature of the gases up to the point of ignition. The combustion of the gases is also in progress during their passage through the smoke flue. The doorway *d* is constructed in the rear wall of the combustion chamber to afford access to its interior for examination, cleaning, and repairs.

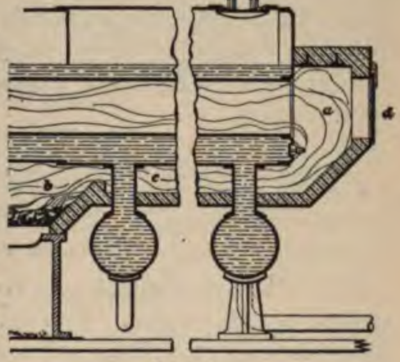


FIG. 59

56. The construction of the form of furnace commonly used with water-tube boilers of the Babcock & Wilcox type is such that there is little opportunity for combustion to take place after the gases leave the firebox. The gases rise nearly vertically from the fuel bed and pass from the firebox immediately into contact with the tubes; the narrow spaces between the tubes divide the gases into thin sheets that are rapidly cooled below the temperature of ignition. The vertical direction of the current of gases in the furnace makes it difficult to secure any considerable admixture of air from the fire-door; the chief dependence for the air supply must, therefore, be on the air that rises through the grates and passes upwards through the bed of fuel. These conditions make it essential that, for complete and economical combustion, a sufficient supply of air be admitted through the grate itself and that the supply be well distributed over the whole grate area so that it may become mixed with the gases almost as

soon as they are formed. It is also important that the grate be placed far enough below the tubes to permit of a thorough mixture of the gases and air and of complete combustion of the gases before they enter the spaces between the tubes. The distance from the grate to the tubes should be regulated in accordance with the volatile contents of the coal; for anthracite or coke, the minimum distance is about 24 inches; for coals containing large quantities of volatile matter and burning with a long flame, a distance of 36 inches or more is often needed.

PASSAGES FOR GASES OF COMBUSTION

BOILER FLUES

57. The longitudinal seams of **boiler flues** may be either riveted or lap welded. Riveted flues are usually made as shown in Fig. 60. They are made in sections, the ends of the sections being fitted one into the other and substantially

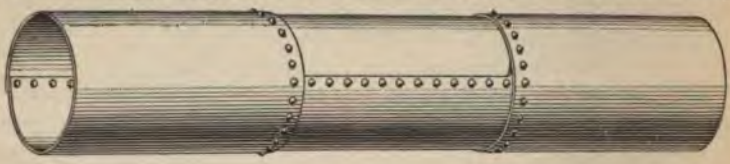


FIG. 60

riveted. Lap-welded flues above 6 inches in diameter and not exceeding 16 inches diameter need not be made in sections, provided that the steam pressure does not exceed 60 pounds per square inch or the length of the flue 18 feet. If the pressure is over 60 pounds and not exceeding 120 pounds, such flues may be made in sections not exceeding 5 feet in length.

BOILER TUBES

58. Materials and Sizes.—The principal purpose of **boiler tubes** is to increase the heating surface of the boiler and thereby increase its steaming capacity. The tubes also divide the water and heated gases into small bodies, thereby

greatly facilitating the transmission of the heat from the gases to the water.

Boiler tubes are made of iron, steel, and brass, and until recently were all lap welded. Lately, however, cold-drawn, seamless, steel tubes have come into extensive use for first-class work. Brass tubes were formerly much used in the Navy, but they have been superseded by steel tubes. Seamless copper or brass tubes not exceeding ¾ inch in diameter are allowed in the construction of water-tube pipe boilers or generators in which liquid fuel is used.

59. The sizes of tubes and flues are designated by their outside diameters to distinguish them from pipes, which are designated by their inside diameters. Tubes more than 5 inches in diameter are usually called flues. The thicknesses of

SIZES OF BOILER TUBES

Outside Diameter Inches	Thickness Fractions of an Inch	Thickness by B. W. G. Number	Outside Diameter Inches	Thickness Fractions of an Inch	Thickness by B. W. G. Number
1	.072	15	4½	.134	10
1¼	.072	15	5	.148	9
1½	.083	14	6	.165	8
1¾	.095	13	7	.165	8
2	.095	13	8	.165	8
2¼	.095	13	9	.180	7
2½	.109	12	10	.203	6
2¾	.109	12	11	.220	5
3	.109	12	12	.229	
3¼	.120	11	13	.238	4
3½	.120	11	14	.248	
3¾	.120	11	15	.259	3
4	.134	10	16	.270	

tubes are expressed in fractions of an inch, and also by wire-gauge numbers. There are numerous wire gauges in use, but the one generally accepted as the standard is the Birmingham

wire gauge, which is usually designated by the capital letters B. W. G.

The thicknesses of boiler tubes and flues, as required by the Board of United States Supervising Inspectors of Steam Vessels, are given in the table in this article.

The table gives the minimum thickness of boiler tubes and flues up to 6 inches in diameter, and of any required length, to carry a working pressure not to exceed 225 pounds per square inch, if deemed safe by the inspectors. Lap-welded flues over 6 inches in diameter and not over 16 inches in diameter, of the thicknesses given in Table I, are allowed a pressure of 60 pounds per square inch if their lengths are not greater than 18 feet. Lap-welded and riveted flues over 6 inches in diameter and not over 16 inches in diameter, and not longer than 18 feet, are allowed a pressure of 120 pounds per square inch if made in sections not over 5 feet long that are properly fitted into each other and substantially riveted.

60. Securing Tubes.—The method of securing the tubes of Scotch boilers is illustrated in Fig. 61, in which *a* is the tube, *b* the front tube-sheet, and *c* the back tube-sheet. The tubes are inserted from the front end of the boiler and driven through both tube-sheets until they project about $\frac{1}{4}$ inch into the back connections, their lengths being such that about $\frac{3}{8}$ to $\frac{1}{2}$ inch will be left outside of the front tube-sheet.

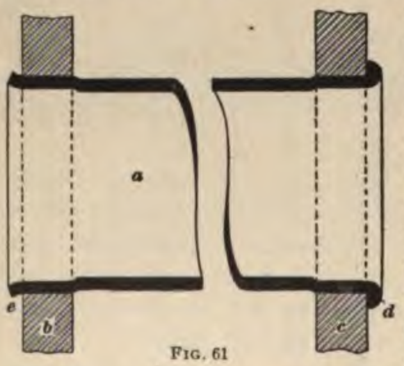


FIG. 61

The ends of the tubes are expanded in the tube-sheets by means of a tool called a *tube expander*, this serving to make the joints tight. After being expanded, the ends of the tubes projecting into the combustion chamber are peened over with a ball-faced hammer and then beaded over the tube-sheet, as shown at *d*, Fig. 61, by a beading tool, commonly called a

boot-tool, made as shown in Fig. 62. The front ends of the tubes are sometimes merely expanded, so that the tubes may be driven inwardly a little and expanded and beaded anew

FIG. 62

should they leak, but more often they are peened over slightly, as shown at *e*, Fig. 61.

61. The **Dudgeon roller-tube expander,** shown in Fig. 63, is most commonly used for expanding boiler tubes. It consists of a body *A* provided with three slots for the reception of the rollers *a*, *a*, *a*. A plate *B*, fastened to the body by the three screws *c*, *c*, *c*, prevents any longitudinal movement of the rollers. The rollers are forced outwards and rotated by the taper pin *C*, which is provided with a head perforated

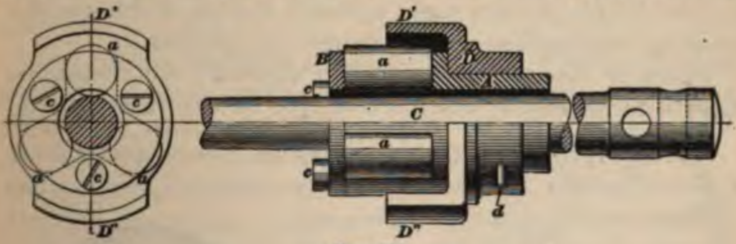


FIG. 63

with two holes at right angles to each other. To make the expander adjustable for different thicknesses of tube-sheets, the hood *D* may be moved longitudinally and may be locked in any desired position by means of a small taper pin *d*, which is driven inwards to lock the hood. The operation of the expander is as follows: The expander is pushed into the tube, the projections D', D'' limiting the depth to which the tool may be inserted. A sharp blow with a copper hammer is struck on the head of the pin *C*, forcing it inwards, and hence the rollers outwards. Next, a bar is inserted into

one of the holes in the head and the pin rotated; the friction between the pin and the rollers causes the latter, and consequently the whole tool, to rotate; this operation expands the tube.

62. The ends of boiler tubes should be annealed before they are expanded; otherwise, they will be hard and brittle and liable to split while being expanded. The process of annealing the tubes is as follows: The ends of the tubes are heated to a red heat in a furnace or forge, and the tubes are then stood on their heated ends in a box containing a mixture of fine charcoal and air-slacked lime, in equal parts, to a depth of 5 or 6 inches, and there allowed to cool. This renders them soft and ductile, which enables them to be expanded without danger of splitting.

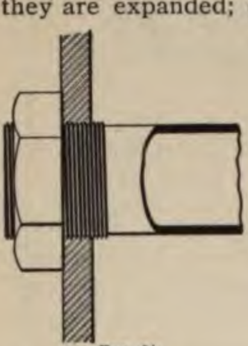


FIG. 64

FIG. 65

Tubes or flues above 5 inches in diameter are commonly riveted to the heads, which are flanged to receive them.

63. **Stay-tubes** are used to stay the tube-sheets, and are tubes of greater thicknesses than the ordinary tubes. The ends are enlarged and threaded, and the tube is screwed into the front and rear tube-sheets. They are secured either by a nut on the outside of the sheet, as shown in Fig. 64, or beaded over, as illustrated in Fig. 65. Sometimes a nut on each side of the sheet is used. From one-third to one-fourth of the tubes in a Scotch boiler are stay-tubes.

FIG. 66

A sectional view of a wrench that is very convenient for screwing and unscrewing stay-tubes is shown in Fig. 66. It

consists of the spindle or mandrel *a*, which is just large enough to enter the tube. Two semicircular grooves *b*, *b* are cut into the sides of the spindle, into which are fitted two short pieces of tool steel *c*, *c*, of the section shown, and held there by any convenient means. Their outer edges being roughened, they grip the inside of the tube whichever way it may be turned.

64. Tubes of water-tube boilers of the Babcock & Wilcox type are secured in a similar manner to those of the fire-tube boiler. The tubes of water-tube boilers of the See and Seabury types are expanded into the steam and water drums, the operation being similar to that practiced with large tubes.

The tubes of the Mississippi boiler are secured in the steam and water drums by screw ferrules. These ferrules are threaded both inside and outside; the inside thread receives the end of the tube, while the outside thread is screwed into the drum. Pipe boilers of the Almy and Roberts types are put together with ordinary threaded pipe fittings.

65. Serve Tube.—A sectional view of a boiler tube that is now being used to a considerable extent in the boilers of modern steamships is illustrated in Fig. 67. It is known as the **Serve tube** and its special feature is that its interior surface is largely composed of a number of longitudinal ribs, as shown at *a*, *a*. The object of these ribs is to increase the amount of surface of the tube that is in contact with the hot gases, and thereby extract a larger amount of heat from them, which is transmitted to the water through the ribs and walls of the tube, thus increasing the efficiency of the boiler.

Fig. 67

66. Spiral Retarder.—It is sometimes the case that in fire-tube boilers with very strong natural draft, or when using forced draft, the gases of combustion enter the stack at a

very high temperature, and much of the heat they contain, which should have been absorbed by the water to make steam, is lost. In order to overcome this difficulty and cause the gases to be retained in contact with the heating surface of the tubes for a longer period of time, and thus utilize more of their heat, an arrangement called a **spiral retarder** is inserted into each tube. The retarder is illustrated in

FIG. 68

Fig. 68, in which *a* is the tube and *b* the retarder, which consists simply of a strip of thin (say, $\frac{1}{8}$ inch thick) sheet iron or steel, the width of which is just equal to the inside diameter of the tube. This metal strip is twisted to a spiral form, very similar to a carpenter's auger, and placed in the tube, wherein it is a loose fit. This spiral strip causes the current of hot gases to take a spiral course through the tube, thereby increasing the distance they have to travel, and keeping them in contact with the walls of the tube a longer period of time, which gives the heat a better opportunity to enter the water.

By gripping the end of the retarder with a pair of blacksmith's tongs, it may be turned around and around, thus loosening any soot or other foreign matter that may have become incrusted on the inside of the tube, after which the retarder can be withdrawn and the loosened incrustation blown from the tube by a steam jet.

MARINE-BOILER ACCESSORIES

CONSTRUCTION, USE, AND CARE

SAFETY VALVES

CONSTRUCTION

1. A **safety valve** is a device used to prevent the steam within the boiler from exceeding a certain pressure. This is accomplished by the steam overcoming the downward force on the valve and opening a passage for the steam to escape into the atmosphere. The valve remains open for some time, reducing the steam to a somewhat lower pressure, and relieving the boiler.

2. A common **lever safety valve**, as applied to boilers of steam vessels, is shown in Fig. 1. The lower end of the casing *A* communicates with the boiler and is bored out to receive the gun-metal bushing *B*. This bushing is secured to the casing, its upper edge *b* being chamfered and forming the seat for the valve *V*. Four arms on the inside of the bushing support the boss *b'*, which is bored out central with the seat, and forms a guide for the lug on the lower face of the valve. Attached to the upper end of the valve is the stem *S*, which passes through a gun-metal bushing in the bonnet *A'* and is guided at its upper end by a bushing in the yoke *Y*. The stem is slotted below the yoke and above

the bonnet, and is provided with knife-edge bearings for the lever *L*, which has its fulcrum at the upper end of the link *l* pivoted at *a*. A weight *W* is placed a certain distance from the fulcrum; this weight acts on the valve stem, and consequently presses the valve to its seat, forming the resistance, or downward force, that the steam acting on the under side of the valve has to overcome. On the steam pressure reaching a certain point, it will balance the downward force, and a slight increase of the steam pressure will cause the valve to lift from its seat, thus forming an annular opening through which the steam escapes into the passage *P* and thence through a pipe (not shown in the figure) into the atmosphere. As the steam pressure becomes less, a point is soon

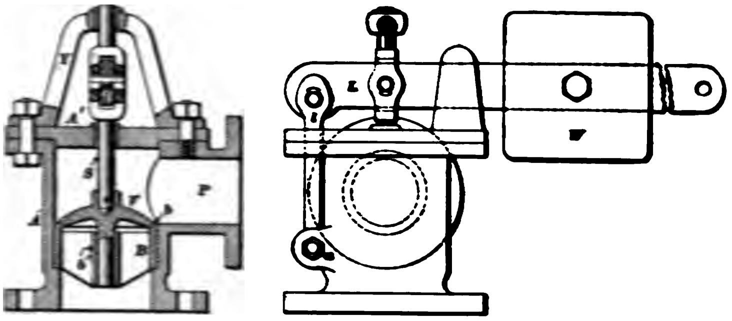


FIG. 1

reached at which the downward force balances the steam pressure, and on the pressure being still further reduced, the valve closes. A small drain pipe should be fitted to the casing just above the valve seat, to carry away any water that may collect, and which, by its weight, would add to the external load on the valve. Lever safety valves are today practically obsolete in sea-going vessels, but are used to some extent on river steamers.

3. **A dead-weight safety valve** is shown in Fig. 2. In this construction, the external load on the valve is formed by cast-iron or leaden weights *W*, *W* piled on a plate *P* attached to the valve stem. The action of this valve, under pressure, is the same as that of the valve just described.

A crown cap, or hood *D*, provided with two rings cast on it and also with a handle, is attached to the upper end of the valve stem *S*, by means of the cotter *C*, and fits the stem loosely. The depth of the slot in the stem *S*, which works in a bushing *F* fixed in the cover, is such that the valve can move upwards a certain distance without lifting the hood *D*, while the hood cannot be raised without raising the valve with it. By means of the forked lever *L*, the valve can be lifted and its freedom of action tested. The handle attached to the hood is for the purpose of turning the valve around on its seat occasionally, thus crushing any scale or other sediment that may lodge on it.

The dead-weight safety valve is the earliest form of safety valve, but is now obsolete in American practice.

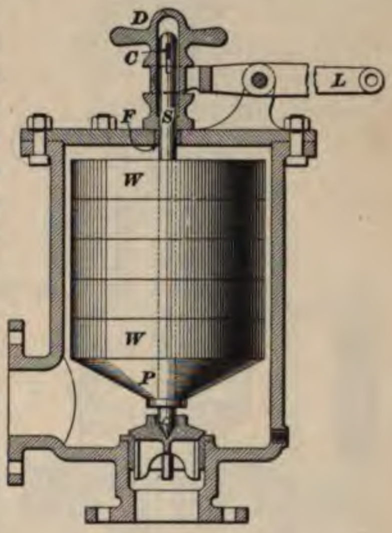


FIG. 2

4. A spring-loaded safety valve is illustrated in Fig. 3. The external load is the force required to compress the coiled spring *T*. By means of the threaded bushing *G*, the tension of the spring can be regulated by screwing the bushing up or down, thus providing a simple method of adjusting the downward force to the pressure at which the valve is to open. The hood *D*, which fits loosely on the upper part of the bonnet or casing *B*, is keyed to the valve stem *S* by means of the cotter *C*, in which is the padlock hole *d*. The hood, and consequently the valve, may be lifted by means of the lever *L*. At the same time, the valve is at liberty to open without lifting the lever. The bushing *G*, which is provided with a locknut *G'*, forms a guide for the

upper end of the valve stem. When the steam pressure lifts the valve off its seat, the spring is compressed; this increases the resistance of the spring and so increases the force acting downwards on the valve. This prevents the valve from lifting any higher, and gives but a limited area for the steam to escape.

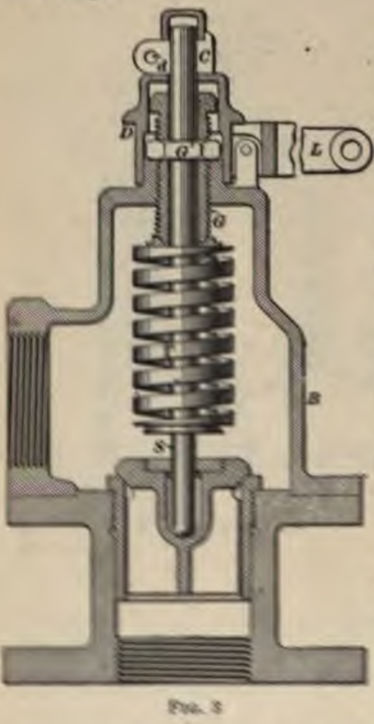


FIG. 4

5. To overcome the increasing resistance of the spring mentioned in Art. **4**, the so-called **pop safety valve** has been designed, and has now superseded all other forms of spring-loaded safety valves; in fact, this type of safety valve is used for most marine boilers. A pop-valve is shown in Fig. 4. A gun-metal bushing *B* is fitted to the lower end of the valve casing *A*. The bushing is threaded and fitted with a movable ring *R* of the cross-section shown in the figure. A threaded bushing *G*, provided with a lock-nut and forming a guide for the valve stem *S*, is used to adjust the external load on the valve. An annular recess *r* is cut into the face of the valve *V*. The other parts *C*, *d*, *D*, *L*, and *T* are the same as in Fig. 3. On the steam pressure lifting the valve, the steam flows through the opening between the valve and its seat and through the annular passage *b-b* into the casing, and thence into the atmosphere. Its free escape being somewhat restricted by the narrow passage *b-b*, a slight pressure accumulates under the face of the valve, and this pressure acting on the increased surface presented causes the valve to open promptly and allows the

steam to escape freely. This additional pressure can be adjusted by raising or lowering the ring *R*, thus reducing or increasing the area of the passage *b b*. The smaller the area of this passage, the faster the pressure will accumulate and the higher the valve will lift; conversely, the larger the area of the passage, the slower will the pressure accumulate, and the longer will be the time required for the full opening of the valve. A pop-valve closes very promptly, thus preventing an undue loss of steam.

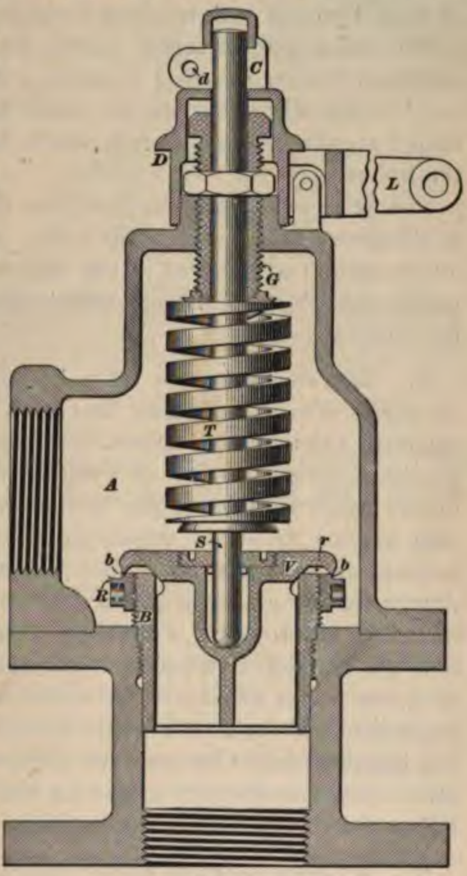


Fig. 4

SIZE AND ARRANGEMENT

6. A safety valve is intended to relieve a boiler of all the surplus steam generated. To accomplish this object, this valve must have a certain area of opening; this area is fixed by the rules and regulations of the Board of Supervising Inspectors of Steam Vessels to be, for lever safety valves, at least 1 square inch to every 2 square feet of grate surface of the boiler to which the valve is attached. For spring-loaded safety valves constructed so as to give an increased lift by the operation of steam after being raised from their

seats, or any spring-loaded safety valve constructed in any other manner so as to give an effective area equal to the first-mentioned spring-loaded valve, the area of opening must be at least 1 square inch to every 3 square feet of grate surface.

All spring-loaded safety valves for water-tube, coil, and sectional boilers required to carry a steam pressure exceeding 175 pounds per square inch must have an area of not less than 1 square inch to every 6 square feet of grate surface of the boiler.

The term *area of opening* refers to the area corresponding to the internal diameter of the valve. The term *effective area* always refers to the area of the annular opening between the valve and its seat, through which the steam escapes when the valve is raised.

7. The safety valves of a battery of boilers must be arranged in such a manner that each boiler shall have one separate safety valve, unless the arrangement is such as to preclude the possibility of shutting off the communication of any boiler with the safety valve or valves employed. This also applies to safety valves provided with an attachment preventing any but an authorized person to increase the downward force on the valve. A valve constructed in such a manner is known as a **lock-up safety valve.** Referring to Figs. 3 and 4, if a padlock were attached to the cotter C at d, the cotter could not be withdrawn unless the padlock were removed first; and, as the hood D covering the adjusting bushing cannot be removed without withdrawing the cotter, no one but the person having the key of the padlock can adjust the compression of the spring.

8. All spring-loaded safety valves must be provided with a lever that will raise the valve from its seat a distance of not less than one-eighth the diameter of the valve opening. The seats of any size of safety valve used on a marine boiler must have an angle of inclination to the center line of 45°. The area of opening of the connection between the safety valve and the boiler must be at least equal in area to the area of the valve.

9. Very often, one lock-up safety valve is provided for every common safety valve. The valves (two or more in number) are usually placed on a separate fitting, shown at *A*, Fig. 5. This fitting may be attached to the shell of the boiler or the shell of the steam drum. It avoids the necessity of cutting a separate hole in the shell for each valve. In the illustration, *B* and *C* are the safety valves, and *b* and *c* their respective escape pipes leading the steam blown off to the main escape pipe. The drain pipes *b'*, *c'* prevent the accumulation of water in the valve casing.

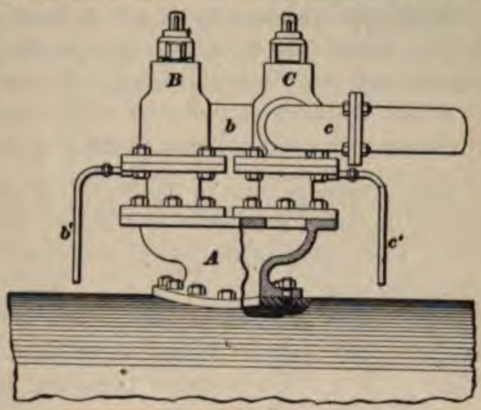


FIG. 5

CALCULATIONS

10. Lever Safety Valves.—No safety valve can open without a slight increase of pressure above that for which it is set; since, in order to lift the valve, the pressure on the under side of the valve, which may be called the *internal*, or *upward*, *force*, must exceed the *external*, or *downward*, *force* on the valve plus the friction of the mechanism of the valve. If the internal and the external forces on the valve are equal, the valve will be in equilibrium (balanced), and an increase of the internal force will cause it to open. A safety valve will not close until the pressure has been reduced somewhat below the pressure at which the valve opened.

11. The point at which a safety valve will blow off depends on the external force on the valve. To be in equilibrium, the external load exerting a downward pressure on the valve must be equal to the internal force exerting an

upward pressure on the under face of the valve. Evidently, the upward pressure is equal to the area of the valve multiplied by the pressure per unit of area.

Whenever the word *pressure* is used in relation to calculations pertaining to safety valves, the **gauge pressure** is meant, unless otherwise stated. By the ***area of a safety valve*** is meant the area of that part which is exposed to the steam pressure when the valve is seated.

12. Suppose that a dead-weight safety valve, as shown in Fig. 2, has a diameter of 4 inches and an external load or force, consisting of the valve and stem, the supporting plate and the weights, equal to 815.8 pounds. It is desired to know the pressure at which the valve will open. Since the internal and external forces must balance, it is evident that $815.8 = 4^2 \times .7854 \times$ steam pressure, in pounds per square inch. Then, $\frac{815.8}{4^2 \times .7854} = 65$ pounds per square inch, nearly, the pressure at which the valve is about to open.

13. In the lever safety valve shown in Fig. 1, the external load depends on the position of the weight W on the lever L. Here the same general law holds good; the external and the internal forces must be equal before the valve is about to open. The internal force, as stated before, is the area of the valve times the steam pressure. The downward force on the valve may be found as follows: Suppose that a weight P, Fig. 6, weighing 100 pounds is

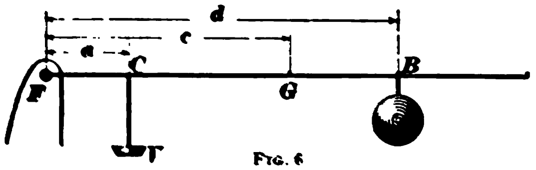


FIG. 6

placed directly on the top of the valve stem C; evidently, the downward force is now equal to the weight of the weight P. Suppose, now, that the weight is removed to the position shown in the figure, the weight's distance d from the fulcrum

being six times greater than the distance *a* from the fulcrum to the center line of the valve. Evidently, the effect of the weight on the valve stem will now be six times greater; that is, the downward force will be $6 \times 100 = 600$ pounds. Hence, to find the downward force, divide the distance *d* by the distance *a* and multiply the quotient by the weight of the weight *P*.

As the valve and stem have a certain weight, the external force is increased an amount equal to that of the weight of the valve and stem, in pounds. Furthermore, the lever has a certain weight, and this, acting at the center of gravity of the lever, adds a certain amount to the downward force. This amount is equal to the product of the distance from the fulcrum of the lever to its center of gravity and the weight of the lever, divided by the distance from the fulcrum to the center line of the valve.

The distance to the center of gravity of the lever may be found by balancing the lever on a knife edge and measuring the distance from the center of the fulcrum to the knife edge. If this should not be feasible, the center of gravity must be found by calculation. In engineers' examinations, the lever is usually given as straight and parallel, in which case the distance from the fulcrum to the center of gravity of the lever should be taken as equal to one-half the length of the lever.

The amount of the downward force on the valve due to the weight of the lever may be found directly by attaching a spring balance by a cord to the lever at the point at which it acts on the valve stem. The spring balance will indicate the correct downward force, in pounds.

Now, to have the valve balance, the area of the valve times the steam pressure (the upward force) must equal the weight times the distance from the fulcrum to the weight divided by the distance from the fulcrum to the center line of the valve; to this downward force must be added the additional downward force due to the weight of the valve, stem, and lever, the sum of these two downward forces constituting the external force.

14. How to find the pressure per square inch at which a safety valve is about to blow off, may best be explained by the following example: Suppose that a safety valve has the following dimensions: The area of the valve is 12.566 square inches; the distance from the fulcrum to the center line of the valve is 4 inches; the weight is 135.2 pounds; the length of the lever, between fulcrum and weight, is 36 inches; the weight of the valve and stem, 9.2 pounds, and the downward force due to the weight of the lever, as found by one of the three methods previously described, 150 pounds. From what has been explained, it should be clear that the valve balances, or is in equilibrium, if the steam pressure $\times$ 12.566 $= 135.2 \times 36 \div 4 + 9.2 + 150$. That is, the steam pressure $\times$ 12.566 = 1,376. Then the steam pressure = $\frac{1,376}{12.566} = 109.5$ pounds per square inch.

Let A = area of valve, in square inches;
D = distance from center line of valve to fulcrum, in inches;
L = distance of weight from fulcrum, in inches;
P = steam pressure, in pounds per square inch;
W = weight, in pounds, of load or weight on lever;
w = weight of valve and stem, in pounds, plus downward pressure due to weight of lever.

Rule.—*To find the pressure at which a safety valve is about to blow off, multiply the weight by the length of the lever and divide this product by the distance from the fulcrum to the center line of the valve. To the quotient, add the downward pressure on the valve due to the weight of the valve, stem, and lever, and divide the sum by the area of the valve.*

Or,
$$P = \frac{\frac{WL}{D} + w}{A}$$

EXAMPLE.—The area of a lever safety valve is 11 square inches; the distance from the center line of valve to the fulcrum, 4½ inches; the distance of the weight, which weighs 125 pounds, from the fulcrum, 35 inches; the weight of valve and stem plus the downward pressure-

due to the weight of the lever equals 137 pounds. Find the pressure per square inch at which the valve is about to open.

SOLUTION.—Applying the rule just given,

$$P = \frac{\frac{125 \times 35}{4.5} + 137}{11} = 100.84 \text{ lb. per sq. in. Ans.}$$

15. To explain how to find where a given weight must be placed on the lever in order that the safety valve may be about to blow off at a given pressure, consider the first paragraph in Art. **14.** In the case cited, the pressure was found to be 109.5 pounds per square inch; hence, the total upward force is $109.5 \times 12.566 = 1{,}375.977$, say 1,376 pounds. This force is partially balanced by the weight of the valve and stem, and the downward force due to the weight of the lever. Consequently, the total upward force $= 1{,}376 - (150 + 9.2) = 1{,}216.8$ pounds, is to be balanced by the downward force. As the downward force is the weight times the length of the lever divided by the distance from the fulcrum to the center line of the valve, it should be plain that the valve will be in equilibrium again if $1{,}216.8 = \frac{135.2 \times \text{the length of lever}}{4}$. That is, $1{,}216.8 = 33.8 \times$ the length of lever, since $135.2 \div 4 = 33.8$. Hence, the lever $= \frac{1{,}216.8}{33.8} = 36$ inches. Or, if $1{,}216.8 = \frac{135.2 \times \text{length of lever}}{4}$, $1{,}216.8 \times 4 = 135.2 \times$ length of lever, and length of lever $= \frac{1{,}216.8 \times 4}{135.2} = 36$ inches.

Rule.—*To find the distance from the fulcrum to the point at which the weight must act, in order to have the valve blow off at a given pressure, subtract the downward force due to the weight of the valve, stem, and lever from the product of the area and the steam pressure. Multiply the remainder by the distance from the fulcrum to the center line of the valve and divide this product by the weight.*

Or,
$$L = \frac{(A P - w) D}{W}$$

EXAMPLE.—At what distance from the fulcrum must a weight of 150 pounds act in order that the valve may be about to blow off at 100 pounds per square inch pressure; the diameter of the valve is $3\frac{3}{4}$ inches; the distance from the fulcrum to the center line of the valve is $4\frac{1}{2}$ inches, and the downward force due to the weight of valve, stem, and lever is 125 pounds?

SOLUTION.—Area of valve = $(3\frac{3}{4})^2 \times .7854 = 11.04$ sq. in. Applying the rule just given,

$$L = \frac{(11.04 \times 100 - 125) \times 4.5}{150} = 29.37 \text{ in. Ans.}$$

16. Suppose it is desired to find the weight that must be placed on a lever to have the valve blow off at a given pressure. Using the example given in Art. **15**, the unbalanced upward force, as previously found, is 1,216.8 pounds. The valve then will balance if $1{,}216.8 = \frac{\text{weight} \times 36}{4}$; that is, if $1{,}216.8 \times 4 = \text{weight} \times 36$; whence, the weight

$$= \frac{1{,}216.8 \times 4}{36} = \frac{1{,}216.8}{9} = 135.2 \text{ pounds.}$$

Rule.—*To find the weight that must act on a lever at a given distance from the fulcrum so that the valve is about to blow off at a given pressure, subtract the downward force due to the weight of the valve, stem, and lever from the product of the area and the steam pressure. Multiply the remainder by the distance from the fulcrum to the center line of the valve, and divide this product by the distance from the fulcrum at which the weight is to act.*

Or,
$$W = \frac{(A\,P - w)D}{L}$$

EXAMPLE.—A safety valve has the following dimensions: Area of the valve, 15.7 square inches; distance of weight from fulcrum, 48 inches; distance from fulcrum to center line of the valve, 5 inches; the downward force due to the weight of the valve, stem, and lever is 182 pounds. Find the weight to blow off at 64 pounds per square inch.

SOLUTION.—Applying the rule just given,

$$W = \frac{(15.7 \times 64 - 182) \times 5}{48} = 85.71 \text{ lb. Ans.}$$

17. Some inspectors of the United States Steamboat Inspection Service prefer to have the lever safety-valve

problems worked out by rules I, II, and III, which follow. These rules will give exactly the same results as the corresponding rules given in Arts. **14**, **15**, and **16**. Their derivations, since it involves a knowledge of algebra, is not given. Rules I and II are colloquially known among American marine engineers as "Roper's rules." When a candidate for marine engineer's license knows that the examining inspector prefers Roper's rules, the candidate is advised to use them.

In the formulas following the rules,

Let A = area of valve, in square inches;
D = distance from center line of valve to fulcrum, in inches;
L = distance of weight from fulcrum, in inches;
P = steam pressure, in pounds per square inch;
W = weight of load or weight on lever, in pounds;
V = weight of valve and stem, in pounds;
w = weight of lever, in pounds;
l = distance from fulcrum to center of gravity of lever, in inches.

The distance from the fulcrum to the weight is usually called the **length of the lever**, but on account of confusing the length from end to end of the lever with this term, it is not used here.

Rule I.—*To find the pressure at which a lever safety valve is about to blow off, multiply the weight of the weight by the distance of the weight from the fulcrum. Multiply the weight of the lever by one-half its length, if the lever is straight and parallel, or by the distance from the fulcrum to the center of gravity, if the lever is tapered. Multiply the weight of valve and stem by the distance from the center line of the valve to the fulcrum. Add these three products together and divide the sum by the product obtained by multiplying the area of the valve by the distance of the center line of the valve from the fulcrum.*

Or,
$$P = \frac{WL + wl + VD}{AD}$$

EXAMPLE 1.—At what pressure will a safety valve having a diameter of 4 inches blow off, when the weight of the valve and stem is 10 pounds; of the lever, 20 pounds; and of the ball, 120 pounds? The total length of the lever, which is straight, is 44 inches; the weight is 40 inches from the fulcrum, and the distance from the center line of the valve to the fulcrum is 4 inches.

SOLUTION.—The area of the valve = $4^2 \times .7854$. As the lever is straight, its distance from the fulcrum to the center of gravity is taken as one-half its length, or $\frac{44}{2}$. Applying the rule,

$$P = \frac{120 \times 40 + 20 \times \frac{44}{2} + 10 \times 4}{4^2 \times .7854 \times 4} = 105 \text{ lb. per sq. in., nearly.} \quad \text{Ans.}$$

Rule II.—*To find the weight necessary to put on a safety-valve lever, multiply the area of the valve by the steam pressure and multiply this product by the distance between the center line of the valve and the fulcrum. Multiply the weight of the lever by one-half its length, if straight and parallel, or by the distance between the center of gravity and the fulcrum, if tapered. Multiply the weight of the valve and stem by the distance between the center line of the valve and the fulcrum. Add the last two products together and subtract their sum from the first product. Divide the remainder by the distance the weight is from the fulcrum.*

Or,
$$W = \frac{APD - (wl + VD)}{L}$$

EXAMPLE 2.—With a safety valve having the dimensions given in example 1, what weight is necessary to have the valve about to blow off at a steam pressure of 100 pounds per square inch?

SOLUTION.—Applying rule II,

$$W = \frac{4^2 \times .7854 \times 100 \times 4 - (20 \times \frac{44}{2} + 10 \times 4)}{40} = 113.66 \text{ lb.} \quad \text{Ans.}$$

Rule III.—*To find at what distance from the fulcrum to place the weight of a lever safety valve, multiply the area of the valve by the steam pressure, and multiply this product by the distance between the center line of the valve and the fulcrum. Multiply the weight of the lever by one-half its length, if straight and parallel, or by the distance of its center of gravity from the fulcrum, if tapered. Multiply the weight of valve and stem by the distance between the center line of the valve and the fulcrum. Add the last two products together and subtract their*

sum from the first product. Divide the remainder by the weight of the weight.

Or,
$$L = \frac{A\,P\,D - (w\,l + V\,D)}{W}$$

EXAMPLE 3.—A safety valve has an area of 11 square inches; the distance from the center line of the valve to the fulcrum is 3 inches; the steam pressure, 40 pounds per square inch; the weight weighs 50 pounds; the lever is straight and parallel, 32 inches long, and weighs 15 pounds; the valve and stem weigh 6 pounds. How far from the fulcrum must the weight be placed?

SOLUTION.—Applying rule III,
$$L = \frac{11 \times 40 \times 3 - (15 \times \frac{32}{2} + 6 \times 3)}{50} = 21.24 \text{ in. Ans.}$$

18. A candidate for American marine engineer's license should thoroughly familiarize himself with the calculations pertaining to a lever safety valve, as a candidate for a marine engineer's license must be rejected by the examining inspectors if he fails to solve safety-valve problems similar to those given in the examples.

19. Spring Safety Valves.—The question often arises, what pressure is a safety-valve steel spring intended for? When made with 13 complete turns, the standard prescribed, the question can be answered by an application of the rule of the Board of Trade, Great Britain, governing this problem.

Rule.—*To find the steam pressure for which a spring is intended, cube the diameter, in inches, of the wire, if round, or the side of square, if square, and multiply by 8,000 for round wire and 11,000 for square wire. Divide the product by the product of the diameter of the spring, in inches, measured from center to center of the wire, and the area of the safety valve.*

Or,
$$P = \frac{d^3\,c}{D\,A},$$

where P = steam pressure, in pounds per square inch;
d = diameter, or side of square, of wire, in inches;
c = 8,000 for round wire and 11,000 for square wire;
D = diameter of spring from center to center of wire;
A = area of safety valve, in square inches.

EXAMPLE.—For what pressure is a spring made of square wire measuring ½ inch and 3 inches in diameter intended, if the valve has an area of 6 square inches?

SOLUTION.—Applying the rule given,

$$P = \frac{.5^3 \times 11{,}000}{3 \times 6} = 76.4 - \text{ lb. per sq. in. Ans.}$$

Spring-loaded and pop safety valves are finally adjusted under pressure by comparison with an accurate steam gauge, increasing or diminishing the tension of the spring until the valve opens at the desired pressure. The rule given will show about what pressure the spring can be used for.

EXAMPLES FOR PRACTICE

1. A dead-weight safety valve having an area of 12 square inches is to be on the point of blowing off at 75 pounds per square inch pressure, absolute; find the weight. Ans. 723.6 lb.

2. What area of safety-valve opening is required for a water-tube boiler carrying steam at 180 pounds per square inch pressure, the grate surface being 48 square feet? Ans. 8 sq. in.

3. At what pressure will a safety valve of the following dimensions blow off: Area of valve, 10 square inches; distance from the valve to the fulcrum, 3 inches; length of lever (the distance from the fulcrum to the point where the weight acts), 30 inches; weight of the weight, 83.1 pounds; weight of valve and stem, 5 pounds; weight of lever, 12 pounds; total length of lever, 32 inches? The lever is straight and parallel. Ans. 90 lb. per sq. in.

4. Suppose all the quantities to remain the same as in example 3, except that the valve is to blow off at 75 pounds per square inch pressure; at what distance from the fulcrum must the weight be placed? Ans. 24.58 in.

5. All quantities remaining the same as in example 3, except that the valve is to blow off at 82 pounds per square inch pressure, find the weight that must be placed on the lever. Ans. 75.1 lb.

6. If the spring of a pop-valve is made of wire $\frac{9}{16}$ inch in diameter and has a center-to-center diameter of 3¼ inches, what pressure is it intended for if used with a valve having an area of 12 square inches? Ans. 50 lb. per sq. in., nearly

USE AND CARE OF SAFETY VALVES

20. See that the safety valve is attached directly to the boiler. If there is a stop-valve between the valve and boiler, have it removed or arranged so that it cannot be shut under any circumstances. Take care that the valve does not become corroded and stick fast to its seat. It is a good plan to frequently lift the valve from the seat and see whether or not it works freely. Do not overload the valve or increase the tension of the spring, and take care that it is not done by others.

In practice, the position of the weight on the lever is usually found by trial in preference to finding it by calculation. Most safety-valve levers are notched and have figures stamped below the notch, which are supposed to represent the pressure per square inch at which the valve will blow off when the weight rests in the notch. However, since it may be possible that the notches have not been correctly located, it is good practice to check the graduation by an actual trial. To do so, get up steam on the boiler, and as soon as the steam gauge shows the blow-off pressure, shift the weight until the valve just commences to blow off. Then fasten or lock the weight, if possible, so that it may not be shifted accidentally. Before adjusting the position of the weight, make sure that all parts of the valve work freely and that the steam gauge is correct.

After adjustment, the valve should occasionally be tested by comparing its blowing-off point with the pressure shown by the steam gauge. If the steam gauge indicates a higher pressure, it shows one of two things: either the steam gauge has become impaired or the valve is out of order. If there is reason to suspect the steam gauge, have it tested. At any rate, however, in order to be on the safe side, the steam gauge may be assumed to be correct and the valve examined to see if everything works freely. If found so, and the weight of a lever safety valve is still at the same mark, it is reasonable to conclude that the gauge is out of order.

It is common practice to connect a pipe to the blow-off side of the safety valve for the purpose of carrying the steam blown off out of the fireroom. Such an escape pipe, while harmless enough when of sufficient area and kept well drained, may become a source of danger if no provision is made for draining it constantly. Instances are not rare when, owing to the absence of a drain pipe, the escape pipe has become filled with water, thus adding greatly to the external force on the valve and rendering it inoperative for the blow-off pressure for which it was set. When an escape pipe is used, it should not be of smaller diameter than the valve, and should have a drain pipe of ample size at its lowest point. No cock or valve should under any circumstances be placed in this drain pipe.

STEAM GAUGES AND WATER GAUGES

STEAM GAUGES

21. Construction.—The **steam gauge** indicates the pressure of the steam contained in the boiler. The most common form is the **Bourdon pressure gauge,** shown in Fig. 7. It consists of a tube *a*, of elliptic cross-section, that is filled with water and connected at *b* with a pipe leading to the boiler. The two ends, at *c*, are closed and are attached to a link *d*, which is, in turn, connected with a rack *e*. This rack gears with a pinion *f* on the index pointer *g*. When the water contained in the elliptic tube is subjected to pressure, the tube tends to take a circular form, and the tube, as a whole, straightens out, throwing the free ends out a distance proportional to the pressure. The movement of the free ends is transmitted to the pointer by the link, rack, and pinion, and the pressure is thus recorded on the graduated dial.

Mercurial gauges, in which a column of mercury was forced into a closed glass tube by the steam pressure compressing the air above the mercury, were formerly used, but were superseded by a gauge of the construction shown in Fig. 7, and which is called a **metallic, or dial, gauge.**

22. Steam gauges are placed where they are immediately within sight of the water tender, but out of reach of rough usage. The steam pipe for the gauge is generally connected to the top of the boiler, sometimes to the steam drum, and, in some instances, the gauge is placed on top of a so-called water column. A siphon should be placed directly under the gauge. This siphon is formed by carrying the steam pipe to a lower level than the gauge and then bending it upwards, thus forming a **U** that is filled with water to prevent the heat of the steam from injuring the spring and the other mechan-

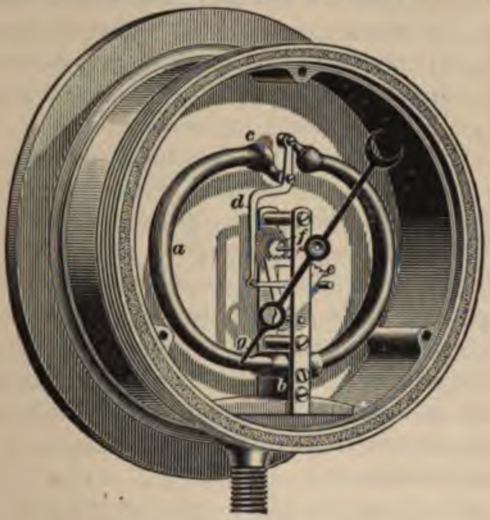


FIG. 7

ism of the gauge, or distorting its action by the expansion of its parts. A small drip cock, shown at *d*, Fig. 8, is fitted to the lowest point of the siphon and serves to let the water out of the leg of the siphon connected with the boiler. If no cock were fitted, the water accumulating in this leg by the condensation of the steam would, by its weight, cause the gauge to indicate a higher pressure than that due to the steam. Care should be taken not to locate the steam-gauge pipe near the main steam outlet of the boiler, since this may cause the gauge to indicate a lower pressure than really exists.

23. Pressure gauges for indicating steam pressure are, in English-speaking countries, invariably graduated to indicate pressure above that of the atmosphere, in pounds per square inch, and show how much the pressure has been increased above the atmospheric pressure. When pressure gauges are used for indicating the pressure in the condenser, they are called **vacuum gauges**, and are invariably graduated to show, in inches of mercury, how much the pressure has been decreased below that of the atmosphere. Then, to find the absolute pressure, the vacuum-gauge reading must be subtracted from 30, and the pressure will be given in inches of mercury. To obtain the absolute pressure in pounds per square inch, multiply the difference between 30 and the gauge reading by .49. Thus, if the vacuum gauge indicates 25 inches, the absolute pressure in the condenser is $(30 - 25) \times .49 = 2.45$ pounds per square inch. The directions just given are entirely correct for normal atmospheric conditions at sea level. Since the pressure of the atmosphere is not constant, but varies between certain limits, it is better, if accuracy is desired, to use the following general rule:

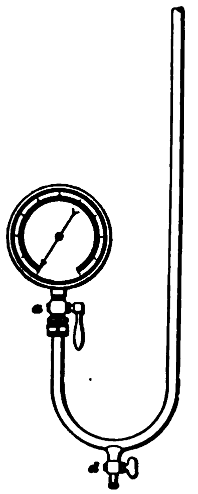


FIG. 8

Rule.—*To find the absolute pressure shown by a vacuum gauge, subtract the vacuum-gauge reading from the reading of the barometer and multiply the difference by .49.*

EXAMPLE.—What is the absolute pressure if the vacuum gauge indicates 19 inches, while the barometer stands at 28 inches?

SOLUTION.—Applying the rule just given,

Absolute pressure $= (29 - 19) \times .49 = 4.9$ lb. Ans.

24. **Compound steam gauges** are occasionally met with, in which the left-hand part of the dial indicates vacuum in inches of mercury and the right-hand part indicates pounds per square inch above the atmospheric pressure. They are

usually found attached to the receivers of multiple-expansion condensing engines.

25. Use and Care of Steam Gauges.—While the engine is running, it will often be noticed that the pointer of the gauge vibrates so much that the pressure cannot be read. This can be prevented by partially closing the cock *a*, Fig. 8. The greatest of care must be taken, however, to prevent an entire closing of the cock. The pointer of a steam gauge will stick occasionally; hence, experienced engineers always jar the gauge a little, in order to dislodge any foreign matter that may be preventing movement of the pointer, before they accept its indication as correct.

Steam gauges will lose their accuracy after they have been in use for some time, owing to the spring losing its elasticity or taking a permanent set. In this case, the gauge will indicate a pressure higher than the actual pressure in the boiler. This can usually be discovered by the pointer failing to return to the zero mark when there is no pressure in the boiler. If the pressure apparently indicated when there is no pressure be subtracted from the pressure indicated when the boiler is under steam, the correct pressure will be given approximately. However, when a gauge shows a wrong pressure, a new one should be immediately substituted and the old one discarded or sent to the maker for repair. When inspecting boilers, the boiler inspector usually tests all steam gauges on board by comparison with an accurate test gauge. The gauge to be tested and the test gauge are both attached to a vessel in which the pressure is raised by means of a small force pump, and the readings of the two gauges are compared at different pressures.

As previously explained, the accuracy of the safety valve can be checked by means of the steam gauge when the latter is known to be accurate. Conversely, when the safety valve is known to be set correctly, the steam gauge can be checked for the blow-off pressure by watching its indication when the valve just blows off. If a steam gauge shows an error of more than 5 pounds, it will be condemned by most boiler

inspectors. Steam gauges should be taken off at least once a month and the connecting pipe cleared by blowing steam through it. When the gauge is off, see that the hole in the nipple is perfectly clear.

Good practice demands that one steam gauge should be attached to each boiler when more than one boiler is used. On some vessels, however, it is not uncommon to see one steam gauge do duty for a whole battery of boilers. This is permissible where several boilers are set in a battery, that is, have a common furnace and are connected by drums at the top and bottom. An arrangement of this kind is never found in sea-going vessels, but is common in vessels navigating the western rivers of the United States of America and many South American rivers.

With some kinds of water, the spring of the steam gauge will corrode. Under no circumstances attempt to fix a corroded spring by soldering up the hole or holes. Instead of this, send the gauge to the maker to have a new spring fitted and adjusted. When replacing the gauge after taking it off, make sure that the valve in the steam-gauge pipe is opened before going further, and then make sure that the gauge is operative. It has happened in numerous instances, in putting up the piping with unions, that the gasket placed between the two parts of the union has been so large that in tightening the nut it has been squeezed out so as to completely stop the hole in the pipe, thus preventing the gauge from showing the pressure.

WATER GAUGES

26. Gauge-Cocks.—A special form of valve or cock attached to the boiler for the purpose of testing the level of the water is known as a **gauge-cock.** The cocks, usually three in number, are placed either on the head or shell or they are attached to the water column. Three gauge-cocks *a*, *b*, *c* are shown in Fig. 9, screwed into the front head of a Scotch boiler. The middle gauge-cock *b* is at the proper water level, generally about 8 or 9 inches above the

top of the combustion chamber. The lower gauge-cock *c* is about 5 inches below the middle cock, and the top gauge-cock *a* is 5 inches above it. Should the top gauge-cock be opened while the boiler is steaming, steam will issue from it. On opening the middle gauge-cock, a mixture of water and steam will issue, and solid water will come out of the lowest cock. When gauge-cocks are fitted in the position shown in the figure, a drip pan *D* is fitted below the cocks to prevent the water from the cocks coming in contact with the head of the boiler and corroding and soiling it. The nozzles of the cocks are pointed in such a direction that the jet of steam or water issuing from one cock cannot strike the one below it and scald the attendant. Gauge-cocks are often placed on a separate fitting, consisting of a tube with its

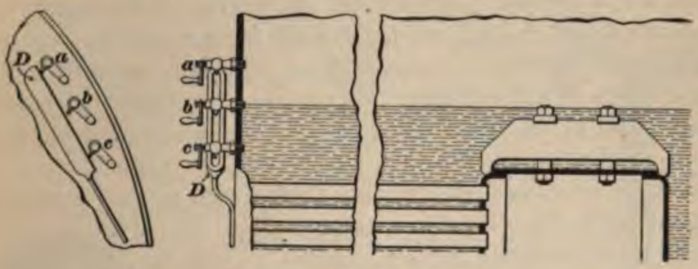


FIG. 9

ends connecting with the steam and water spaces of the boiler, sufficiently below and above the water level to be out of reach of the violent ebullition going on at the surface of the water. If the cocks are connected directly to the head or the side of the boiler, this violent ebullition may cause the gauge-cocks to indicate a wrong water level.

27. Gauge-cocks are made in a great variety of forms, and it is largely a matter of choice which to adopt. Those cocks shown at *a*, *b*, *c*, Fig. 9, belong to a type that is used extensively. It consists of a threaded spindle or stem having a small conical disk valve on its inside end, which fits a seat inside the cock body, and has a crank-shaped handle on its outside end. The cock is opened by revolving the

crank-shaped handle about a quarter of a turn. This cock is suitable for boilers of large diameters on which the cocks are placed too high to be conveniently reached by hand from the fireroom floor. When they are so placed, they are usually operated by a rod having a hand grip at one end and an eye made to fit over the handle of the cock at the other end.

FIG. 10

Another form of gauge-cock is shown in Fig. 10. It consists of the body *a*, the stem *b*, to which the valve is attached, and the nozzle *c*. This cock is similar in construction to the one just described, with the exception that instead of having a crank-shaped handle, it has a disk of hardwood for turning the valve. The wood, being a non-conductor of heat, prevents the hand from being burned. This cock is suitable for small boilers only, on which the cocks can be reached by the hand of the operator.

A form of gauge-cock that has come into extensive use is shown in Fig. 11. It is known to the trade as the *register pattern gauge-cock* and consists of the body *a* and a nozzle, which consitutes the valve seat. The weighted handle *b*, called the *ball*, is secured to the body of the cock by the pivot *c*. A strip of sheet rubber *d* is inserted in a slot in the base of the handle, which makes a water-tight joint on the nozzle when the cock is closed. The cock is kept closed by the weight of the ball. To open the cock, it is only necessary to lift the ball, and it will close itself when the ball is released. When the vessel gives a sudden lurch, this cock is liable to spring open for an instant and blow out some hot water on the heads of the boiler

FIG. 11

attendants; this is a serious disadvantage, but not serious enough to warrant the exclusion of this kind of cock from marine work.

Still another gauge-cock is illustrated in Fig. 12. It is

FIG. 12

known as the **Mississippi gauge-cock** and consists of the body *a* and the rod or spindle *b*. This rod has the valve *c* on one end and the push handle *d* on the other end. The valve fits into a seat at the end of the cock body. The steam or water flows out through the orifice *e* into the atmosphere when the gauge is opened. This is done by pushing against the handle with the hand, and on the handle being released, the pressure of the steam or water on the back of the valve closes it and keeps it closed until opened again.

The above-described cocks are either screwed directly into the head of the boiler or else into a separate fitting.

FIG. 13

28. Glass Water Gauge.—A glass tube whose lower end communicates with the water space of the boiler and whose upper end is in communication with the steam space, is known as a **glass water gauge.** If in good order, the level of the water in the gauge will be the same as in the boiler. Boilers, in good practice, are provided with both cocks and gauges. Fig. 13 shows a common form of gauge-glass connection. The lower fitting connects with the water space, and the upper fitting with the steam space, of the boiler. A drip cock is placed at the lower end of the glass for the purpose of draining it. The

fittings may be screwed directly into the boiler head. The gauge should be so located that the water will show in the middle of the gauge glass when at its proper level in the boiler. The fittings are provided with valves for the purpose of shutting off the gauge from the boiler when a glass tube breaks and it becomes necessary to put in a new one. The valves are also used for blowing out and testing the working of the gauge.

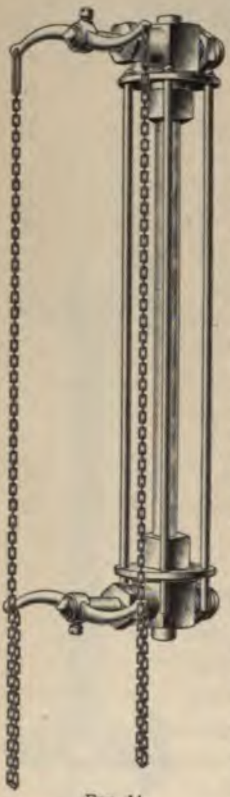


FIG. 14

29. It frequently happens that the glass tubes of water gauges are broken. This is usually the result of improper fitting of the tube; thus, the tube may be too long, so that when expanded by the heat it will be crushed between the fittings. Being improperly packed is also a fruitful source of the breakage of glass tubes. They may be packed too tightly or not properly centered at the ends. When the glass tube of a water gauge breaks, a considerable amount of steam and hot water is blown from it and the water showered about the fireroom. To obviate this in a measure, the quick-closing water gauge, illustrated in Fig. 14, was introduced. The two valves are provided with yokes that are connected together by chains, the bights of which hang down into the fireroom. One pull on the chain will close both valves. This may be done very quickly, but even then a man may be severely scalded before he can pull the chain.

30. To overcome the difficulty referred to in Art. **29**, self-closing glass water gauges were designed, one form of which is illustrated in Fig. 15. The principal feature is the two balls in the steam and water passages, as shown in the sectional portion of the figure. These balls are enclosed in

conical chambers provided for them in the upper and lower connections of the gauge to the boiler. Under normal conditions, the balls lie passively in the large ends of the chambers, there being sufficient space between the balls and the walls of the chambers for the steam and water to flow to the glass tube unobstructed and without creating enough current to move the balls. But, should the glass tube break, there will be a sudden rush of steam or water through the passages, which will force the balls into the small ends of the chambers and close the openings communicating with the glass tube. After the balls close the openings, the pressure of steam and water behind them would hold them in these positions until the pressure in the boiler was blown off if some means were not provided to force them away from the openings after they have served their purpose. This is accomplished by having a small projection, or teat, on the under side of the valve, which pushes the ball from the opening when the valve is screwed shut; the ball will then roll, by gravity, to the large end of the conical chamber, where it will remain until it is required for service again by the breaking of another glass tube. The balls are prevented from rolling back into the boiler by wire pins driven across the passages behind the balls. To preserve the self-closing feature of this gauge, the valves should be screwed back as far as they will go while the steam pressure is on, so as to allow the balls to act promptly when occasion requires.

FIG. 15

31. A glass water-gauge tube may be cut to the proper length by the small tool illustrated in Fig. 16, and called a **glass-tube cutter.** It consists of the metal rod *a* having the thumb and finger handle *b* at one end and the revolving

disk or wheel *c* at the other end. The wheel is made of hardened tool steel and is provided with a knife edge. The guide *d* regulates the amount of the tube to be cut off and is held by the thumbscrew *e*. The proper length of the tube is found by measuring the distance between the fittings, allowing a small margin for expansion. The guide is then slid to the proper position to cut the tube the required length, and is secured there by the thumbscrew. The tool is now inserted into the tube with the cutting edge of the wheel against the inside surface of the tube and turned around by the thumb and fingers. By so doing, the wheel will score the inside of the tube, after which the tool is removed from the tube and the edge of the part of the tube to be detached is inserted into the slot *f*; then, by a sharp downward movement of the handle end of the tool, the tube will be broken off squarely where scored. If a cutting tool is not at hand,

FIG. 16

a glass tube may be cut by filing a groove around it with a sharp-edged file and then tapping it sharply with the file. A little turpentine applied to the file will facilitate the filing operation. Another way to cut a glass tube is to tie a turn or two of lamp wicking or soft string, saturated with turpentine or other inflammable liquid, around the tube at the place where it is desired to cut it, and set the wicking or string on fire; then, *while it is hot*, strike the end of the tube a quick, sharp blow with a piece of metal, when the tube will generally break at the place where it was heated by the burning string.

32. Water Columns.—Gauge-cocks and glass water gauges connected directly to the boiler head are open to the objection that the violent ebullition at the surface of the water will cause them to indicate a wrong water level. To overcome this objection, they are frequently placed on a separate fitting known as a **water column**, which consists

of a large tube with its ends connected to the steam and water spaces of the boiler far enough above and below the water level to be out of reach of the violent ebullition of the surface of the water.

A water column is shown in Fig. 17. The top and bottom ends of the column *A*, which is simply a cast-iron tube, are connected to the steam and water spaces of the boiler by the pipes *B* and *C*, respectively. The gauge-cocks *a*, *a*, *a* and the steam gauge *S* are fitted to this column. It will be noticed that a coil siphon is used below the steam gauge. No drip cock needs to be fitted to this kind of siphon, as no water can collect and disturb the indication of the gauge. The construction of the glass water gauge is as follows: Two fittings *b*, *b*, which are provided with cocks *e*, *e*, are screwed into the water column. These fittings connect with the top and bottom of a glass tube *f* open at both ends. Each fitting is provided with a stuffingbox *d* to make a steam- and water-tight joint between the fitting and the glass tube. To the lower fitting, a drain cock *g*, provided with a waste pipe, is attached; this cock is used to empty the glass tube. The water column is attached to the boiler at such a height that the water, when at its proper level, will show in the middle of the glass tube. It will be seen that the water level in this case is self-indicating, and always in plain sight of the

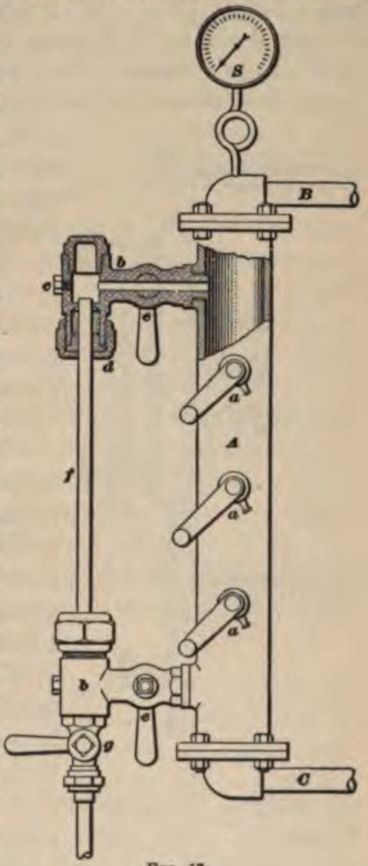


FIG. 17

attendant. The communications of the tube with the steam and water may be shut off by means of the cocks e, e in case the tube breaks.

Another form of water column is illustrated in Fig. 18. This column is more extensively used in America than the one just described. The principles governing both, however, are the same, the difference being only in the details. The body of the column is shown at a, the upper and lower glass-gauge fittings at b, b, the gauge-cocks at c, c, c, and the drip pipe, containing a valve, at d. It will be observed that screw plugs are fitted opposite the ends of the pipes connecting the column with the boiler, for the purpose of cleaning out those pipes with a wire should they become choked. There is also a drip cock placed below the glass tube.

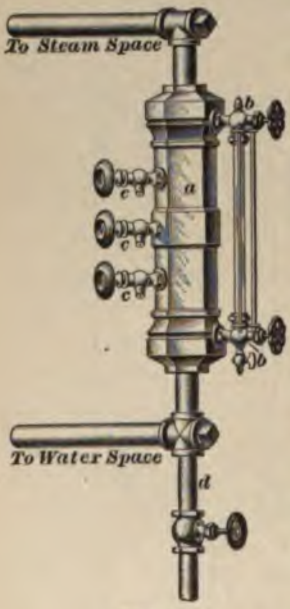


FIG. 18

33. Use and Care of Glass Water Gauges and Water Columns.—Too much reliance must not be placed on the gauge glass. If the water is muddy or contains soda, it is liable to foam, and the glass cannot give the true water level. Again, the connections between the glass and boiler may become so filled with incrustation that scarcely any water can enter the gauge. To prevent this, the glass should be blown out frequently. The water gauge and water column should be tested at least once each watch; i. e., every 4 hours. When the water gauge is attached directly to the head, open the drain cock to blow out the glass. Observe if the water returns immediately to its former level when the drain cock is closed. If it fails to do so, this indicates that the lower fitting is choked with sediment or scale. Should the water

fail to leave the glass, or leave it very slowly, it indicates that the upper fitting is choked. When this test shows the gauge to be out of order, it should be repaired at the first possible opportunity, running in the meantime by the gauge-cocks. To remove all temptation to look at the glass, cover it with any material handy. While this may seem an unnecessary precaution, it may be the means of preventing an explosion.

34. When water columns are used, they usually have a valve in each connecting pipe. To test both the gauge and the column connecting pipes at the same time, *double shut off* one connection and see if the proper fluid comes through both drain cocks. That is, to test the water connection, shut the upper valve of the gauge glass and the valve in the steam connection. Then open, successively, the drain cocks of the glass water gauge and water column. If water issues in a constant stream from both, the water connection is clear. Now open the upper valve of the gauge glass and the valve in the steam connection and close the lower gauge-glass valve and the valve in the water connection. If steam flows freely from both the gauge-glass and the water-column drain cocks, the steam passages and the column itself are clear. Close the drain cocks again and *open all valves.*

This method of testing is commonly expressed in a somewhat ungrammatical form as follows: *Double shut off what you get, and see if you get the other.*

35. When the gauge-cocks and glass water gauge are attached to a water column, a test of the glass water gauge may show it to be unreliable. Then, in order to run by the gauge-cocks, it is first necessary to test and prove that the water column is in good order. To test the water column, shut off the glass water gauge on top and bottom; then shut the valve in the steam connection and open the water-column drain, from which water should flow in a full stream, the valves in the water connection being open. Now close the valve in the water connection and open the valve in the steam connection, when steam should issue freely from the water-column drain.

36. When the water column has no valves in its steam and water connections, it cannot be properly tested. Opening the drain cock of the column will merely prove the water connection to be clear, by water issuing in a solid stream, but will not prove either the steam connection or the column to be clear. Hence, it is advisable to have valves in the steam and water connections of the column. If these valves are not fitted, the glass water gauge should be tested as though it were directly attached to the boiler; if the gauge does not prove right, it is best to assume that both the water column and the glass water gauge are out of order, and to shut down until they can be repaired, unless gauge-cocks are fitted separately to the boiler, by which to run until repairs can be effected.

37. The practice of testing the water column only cannot be too severely condemned, because the testing of the column does not prove the glass water gauge to be correct. When possible, test the glass water gauge and the column separately and prove both to be correct. To prevent the water column from choking up, drain it frequently, and do the same with the glass water gauge. Always supplement the draining by the test given. The water gauges are the most important accessories to a steam boiler, and too much care cannot be bestowed on having them absolutely reliable.

38. The pipes for the water column should run as straight as possible and connect directly to the boiler. Under no circumstances whatsoever should these connecting pipes be used for any other purpose or have any other pipe connection. When observing the water gauge while the boiler is under steam, note particularly whether the water showing in the glass is stationary or not. If the water level does not fluctuate, or pulsate up and down, as it were, it is an infallible sign that the glass water gauge or the water column is out of order. Immediately test the glass water gauge and water column, and if draining fails to clear them, shut down the boiler until they can be cleared, unless the boiler has gauge-cocks directly attached by which to run until repairs can be made.

39. Float Water Gauge.—The feedwater used in the boilers of vessels navigating the western and southern rivers of the United States of America is usually quite muddy, and as the boilers are forced considerably, a good deal of foaming is produced. The ordinary glass water gauge and gauge-cocks will not show the true water level when the boiler is foaming, and as this is taking place more or less all the time with the bad feedwater used and the high rate of evaporation, it is very desirable to have a water-level indicator that will not be affected by foaming. Hence, the **float water gauge** has been designed, and is, in variously modified forms, in common use on the boilers of river

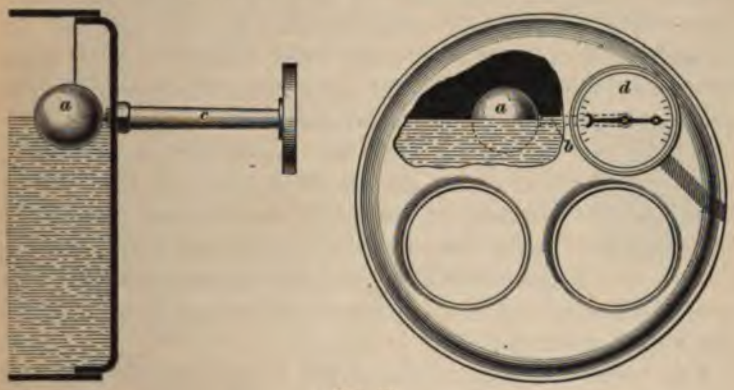


FIG. 19

steamers, taking the place of the glass water gauge. Fig. 19 is an illustration of such a gauge; *a* is a hollow copper sphere, or float, fastened to one end of the lever *b*, which is rigidly fastened to a spindle free to turn in the dial stem *c*. This stem carries a large dial *d*, graduated as shown. A pointer is fastened to the spindle, and moving in front of the dial, indicates on it the height of the water in the boiler. Now, as the float cannot float on foam, this gauge will not be affected by foaming, but will always indicate the true water level, provided that the spindle moves freely. To make a steam-tight joint between the spindle and boiler, a stuffingbox and gland is employed, and great care must be

taken, in packing it, not to pack it too tight. The float must be free to move with any variation of the water level; if the spindle is packed too tight, it may stick and indicate plenty of water in the boiler when the water really is dangerously low.

40. Fusible Plugs.—A safety device having a part that fuses at a low temperature, and is known as a **fusible plug,** is attached to most marine boilers. The purpose of a fusible plug is to give warning in case the water in a boiler should become low. The plug consists of a bronze casing, as shown in Fig. 20, hollowed out, tapering, and threaded on the outside to screw into the boiler plate or tube. The hollowed-out portion of the plug must be filled with Banca tin, which melts at a temperature of about 443° F. As long as the plug is well covered with water, the fusible metal is kept from melting by the comparative coolness of the water; but should the water sink low enough to uncover the top of the plug, the filling quickly melts and allows the steam to rush out, thus giving warning of the shortness of water. The plug shown has a conical filling, the larger end of the filling receiving the steam pressure. The conical form of the filling prevents its being blown out by the steam pressure.

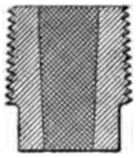


FIG. 20

The rules and regulations of the United States Board of Supervising Inspectors of Steam Vessels require that all steamers plying on United States waters or sailing from United States ports shall have inserted in their boilers plugs of Banca tin at least $\frac{1}{2}$ inch in diameter at the smallest end of the internal opening, in the following manner: Cylindrical boilers with flues shall have one plug inserted in one flue of each boiler; and also one plug inserted in the shell of each boiler from the inside immediately below the fire-line, and not less than 4 feet from the forward end of the boiler. All firebox boilers shall have one plug inserted in the crown of the back connection or in the highest fire-service of the boiler. All upright tubular boilers used for marine purposes shall have a fusible plug inserted in one of the tubes at a point

at least 2 inches below the lower gauge-cock, and said plug may be placed in the upper head-sheet when deemed advisable by the local inspectors. The bronze casing of all fusible plugs, unless otherwise provided, shall have an external diameter of not less than that of a $\frac{3}{4}$-inch gas- or steam-pipe screw tap; except, when such plugs shall be used in the tubes of upright boilers, plugs may be used with an external diameter of not less than that of a $\frac{3}{8}$-inch gas- or steam-pipe screw tap, and having the smallest end of the opening in the plug at least $\frac{1}{4}$ inch in diameter.

41. As with other safety devices, dependence can only be placed on a fusible plug when it is given intelligent and reasonable care. It should be removed frequently and examined to see that the filling is not covered by hard scale. Instances are not rare when the filling has melted out and the steam been prevented from issuing by a heavy covering of scale. The filling should be renewed at least once a year. Before screwing the plug home, smear a liberal supply of plumbago (graphite) on the threads; this will allow the plug to be easily removed. Do not use any oil; this will become carburized, owing to the high temperature, and will make it quite difficult to remove the plug.

For the filling of fusible plugs for American marine boilers, pure Banca tin must be used. This can be procured almost anywhere, although, in general, it is cheaper to keep a small supply of plugs on hand, and simply replace the plug with a new one, instead of refilling it.

When taking charge of an old boiler, it is well to examine the fusible plug to see if it really is what it is supposed to be. Instances are not rare when ignorant persons have either replaced the fusible plug by a solid gas plug or stopped up the hole in the plug by driving wood or iron into it after the filling had melted out.

BLOW-OFF APPARATUS

BOTTOM BLOW-OFF

42. In order that a boiler may be emptied either partially or entirely, a cock called the **bottom blow-off cock** is attached by a short pipe to the lowest point of the boiler and is connected with a waste pipe discharging overboard. Fig. 21 shows the manner in which this cock is often constructed. The shell *A* of the cock is provided with two rectangular passages *B*, *B'*, and has fitted to it a taper plug *P*, pierced by the trapezoidal hole *b*. The upper end of this plug is cylindrical and somewhat smaller in diameter than the shell, thus leaving a space for the insertion of packing. With the plug in the position shown, communication between the passages *B* and *B'* is shut off; but on the cock being given a quarter turn, the hole *b* in the plug will be brought in line with the passages *B* and *B'*, thus allowing the steam or liquid in one passage to pass to the other, and from thence to its destination. A cock in which the passages are directly opposite, like in the one shown, is called a **straightway cock.** If the passages are at an angle to each other, the cock is called an **angle cock.** The upper end of the gland *G* forms a cap, which is bored out large enough to admit the body of the spanner, or key, *K*, and is provided with a slot *S*, shown in the plan view, in such a position that in removing the spanner from the square end of the plug, the tongue *t* of the spanner cannot pass through the slot *S* unless the cock is fully closed. It is the custom in American steam vessels

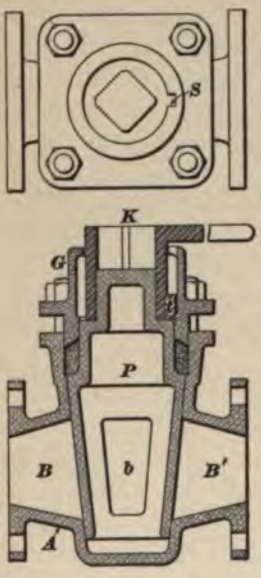


FIG. 21

to put this spanner in the care of the engineer on watch or in charge, thus preventing any person but himself, or a person authorized by him, from opening the blow-off cock. After closing the cock, the spanner is removed and returned to the engineer. Blowing off with the bottom blow is sometimes called *blowing down*.

Although globe valves are extensively used on blow-off pipes, they are objectionable for the reason that, though tightly screwed down, the valve may not be properly closed on account of a piece of scale or similar matter getting between the valve and its seat. As a result, the water may leak out of the boiler unperceived. Formerly, brass plug cocks were used almost entirely, which, owing to their sticking tightly, were superseded by globe valves and gate valves. Within the last few years plug cocks packed with asbestos have been placed in the market, the asbestos packing removing the objectionable features of the plug cock. Many engineers now insist on the use of these cocks for the blow-off pipe. Gate valves are also, to some extent, open to the same objection as globe valves.

The waste pipe, or blow-off pipe, connects with an angle cock attached to the side of the vessel. The construction of this cock is similar to the one attached to the boiler. When it is desired to blow off the boiler, this cock is opened first, and then the cock attached to the boiler; this operation is reversed when the boiler has been emptied or blown out sufficiently. Should the waste pipe break while the boiler is blowing off, or at any other time, the water from the sea would flow into the ship if no cock were fitted to the ship's side. It will thus be seen that the fitting of this cock is merely a precautionary measure.

SURFACE BLOW-OFF

43. Marine boilers usually have a pipe and cock or valve attached near the water level, the pipe extending inside of the boiler and terminating in a scoop or ladle, called a **scum pan,** placed about 3 inches below the water level. The

cock itself is known as the **surface blow-off cock,** or **scum cock,** and has a waste pipe connected usually with the bottom blow-off pipe, between the bottom blow-off cock and the cock on the side of the vessel.

The surface blow-off cock is used to remove the grease and oil and other impurities floating on the surface of the water. The ladle at the end of the pipe serves merely as a funnel for the collection of these impurities. Sometimes a trough is fitted, running across or from the front to the back of the boilers, having the surface blow-off connected to the bottom of the trough.

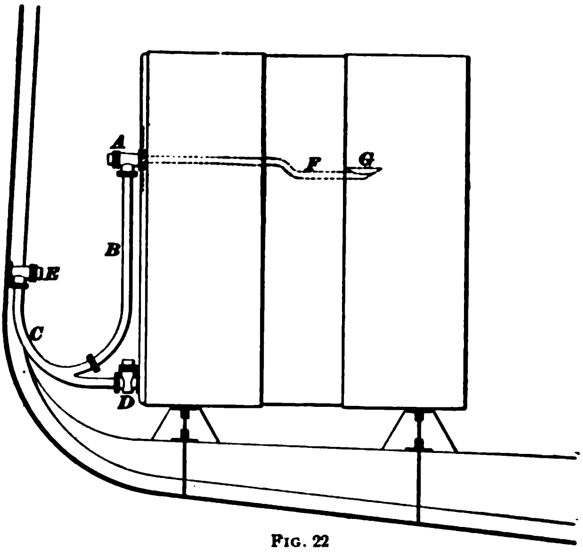


FIG. 22

The ordinary arrangement of the blow-off apparatus for a Scotch boiler is shown in Fig. 22. At *A*, the surface blow-off cock is shown; its waste pipe *B* connects to the waste pipe *C* of the bottom blow-off cock *D*. The water discharges overboard through the cock *E*. The internal surface blow-off pipe is shown at *F*, and scum pan at *G*.

44. After either blow-off has been used for the purpose of partially emptying the boiler of water, the greatest care should be used to make sure that the cocks or valves are

completely closed. If there is a leak, it may be discovered by feeling the blow-off pipe at some distance from the boiler. If the pipe is very hot, it is an indication that the blow-off valves or cocks are leaking, provided that the test is made long enough after the blowing down to give the pipe a chance to cool.

PIPE FITTINGS

VALVES

45. For the purpose of controlling the flow of fluids through pipes, valves and cocks are universally used. **Valves** that allow the fluid to flow through them in either direction are divided into two general classes, viz., *globe valves* and *gate valves*.

46. **Globe valves** are made in a variety of forms, following the same general idea of construction. A common form

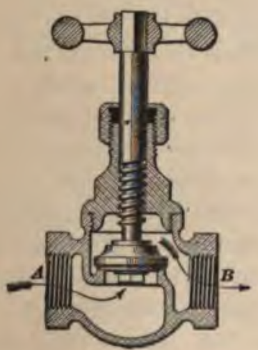


FIG. 23

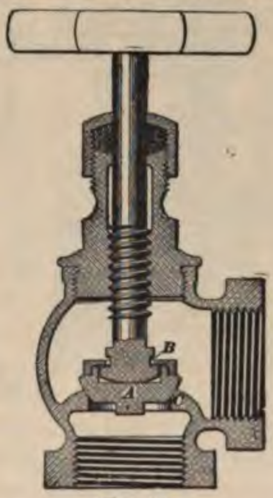


FIG. 24

is shown in Fig. 23. The fluid enters at A and flows out at B. The opening in the valve seat is closed by a flat removable disk, which may be renewed when worn so as to leak. Another common construction is shown in Fig. 24.

This style of valve is used at the junction of two pipes at a right angle, and hence is termed an *angle valve*. The seat of the particular valve shown is beveled; when worn, it may be made tight again by grinding it in. Globe valves should be attached to the pipes in such a manner that the valve will close against the pressure. This will allow the valve stem to be packed without blowing off the steam from the boilers.

47. The steam generated in the boiler is carried to its destination by the steam pipe. This pipe is provided with a stop-valve, which, for large sizes of pipes, is a modification of the globe angle valve shown in Fig. 24. The usual construction of a large stop-valve is illustrated in Fig. 25. A cast-iron casing or body *A* is bolted to a nozzle riveted to the boiler or steam drum. The lower part of the casing is fitted with the gun-metal ring *B*, forming the seat for the valve *V*, which is guided by wings, as shown in the figure. The hand wheel *H* is fixed on the end of the valve stem *S*, which works inside a brass nut screwed into the yoke of the bonnet or cover *C*; the valve stem passes through the stuffing-box (with its gland and neck ring as shown), and is so attached to the valve *V* that the latter can be raised or lowered without turning around when the valve stem is revolved by the wheel *H*. To prevent any movement of the brass nut in the yoke, it is locked in position by means of a small pin shown at *D*, which passes through the yoke and nut. The steam pipe is bolted to the flange *F*. When the valve is raised, the steam flows through the annular

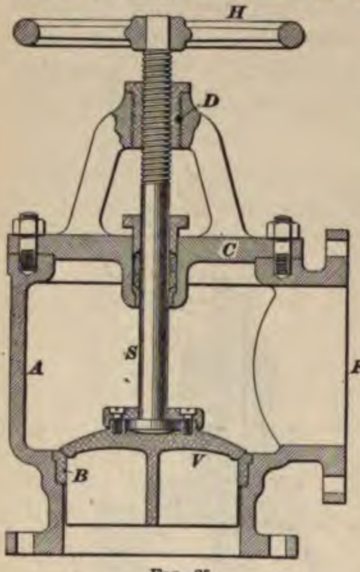


FIG. 25

opening between the valve and the seat, and thence into the steam pipe.

48. The waterway through a globe valve is so contorted that it obstructs the flow of a fluid through the valve to some extent. To overcome this objection, **gate valves** have been designed, a common form of which is shown in Fig. 26. By turning the stem *B*, the wedge-shaped disks *A*, A_1 are moved across the seats *c*, c_1, and the orifice is opened or closed gradually. The disk A_1 has cast on its lower side a projection *D* that rests on a corresponding projection *E* that is cast with the valve body. These two projections form a stop for the disk A_1; when it has come to a stop, a further turning of the stem wedges the two disks apart, pressing them tightly against the seats. A gate valve may be put on to receive the pressure on either side.

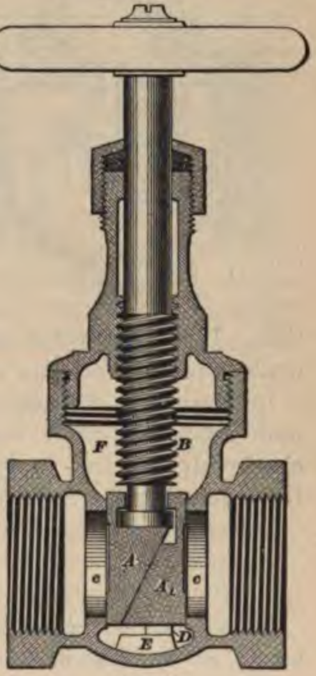


FIG. 26

49. Check-valves are valves designed to permit the flow of fluids in one direction only and to positively prevent any return flow. The most common form of check-valve is that known as a **globe check.** It is shown in Fig. 27. The valve *A* is a solid disk of metal ground to the beveled seat *B*. It is guided by the wings *C* and *E* above and below the seat. The fluid passes in the direction of the arrows.

An improved form of check-valve, known as a *swing check*, is shown in Fig. 28. The valve disk is attached to an arm that swings on a pin, as shown. The passage of the fluid through this valve is more direct than in the globe check. The fluid passes through the check in the direction shown

by the arrows, that is, from *A* toward *B*. In case of a rapid flow, the projection *C* on the end of the arm to which the valve is attached strikes against the bottom of the screw *D*, and is thus kept from going too far.

The two kinds of check-valves illustrated are not very well adapted for working in any other except a horizontal posi-

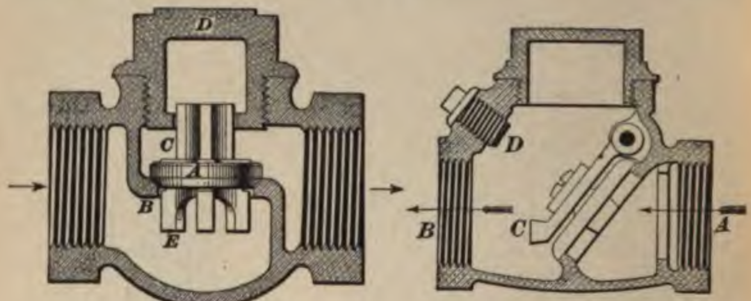


FIG. 27 FIG. 28

tion. If a check-valve must be used in a vertical pipe, one made especially for this purpose should be obtained.

In marine work, when several boilers are fed by a common apparatus, it is customary to fit each boiler with a check-valve having an adjustable lift, and to use this valve for regulating the amount of feedwater entering each boiler.

COCKS

50. For controlling the flow of liquids, **cocks** are often used. In marine work, they are chiefly used on blow-off pipes, for the water service of the engine room and fireroom, and on the pipe lines used for fire-service. Ordinary water cocks made with a brass body and a brass plug ground into it are not well adapted for blow-off cocks; special blow-off cocks are made and can be obtained from all reputable makers. The objection to the ordinary plug cock is its tendency to leak around the bottom and the difficulty of moving the plug after the cock has been closed for some time. To overcome this difficulty, asbestos-packed plug

ocks have been designed and are gradually coming into xtensive use. These cocks have dovetail grooves cast into he body, into which asbestos is tightly driven. The sbestos, being slightly elastic, fits snugly against the plug, hus making a tight joint; at the same time, owing to the mall amount of friction, it allows the plug to be turned easily. Since the asbestos is not affected by heat or moisture, it is quite durable.

The use of ground plug cocks should be avoided as much as possible in marine work; especially the use of large-sized cocks of this pattern. They cannot be kept tight for any length of time if used much, are difficult to repair, and are liable to stick fast in the body when only used occasionally. An attempt to move them when stuck will often result in twisting off the plug end. They answer fairly well in very small pipe lines, say up to $\frac{3}{4}$-inch nominal pipe diameter, but for all larger pipes, valves or asbestos-packed cocks are to be preferred. As a general rule, they should not be used for steam pipes, although their use is customary on indicator piping and steam-gauge piping.

EXPANSION JOINTS

51. Where more than one boiler is used on a steam vessel, all boilers are connected by short branch steam pipes to one large pipe called the *main steam pipe*. Each boiler is provided with a stop-valve, by means of which the communication of the boiler with the main steam pipe may be shut off, if desired. The main steam pipe is also provided with a stop-valve close to the engine, for the purpose of shutting off all communication of the steam with the engine. As the heat of the steam expands the metal of which the main steam pipe is composed, and consequently increases its length, means must be provided by which this increase in length may be taken care of. This is usually accomplished by means of expansion joints, one form of which, known both as a **packed expansion joint** and a **slip joint**, is

shown in Fig. 29. It has a body *A* into which the brass bushing *B* is forced. Into the latter is fitted a sliding tube *C*. Packing is placed in the stuffingbox *D*, and is held in position by the gland *E*, the gland being screwed down by means of bolts, one of which is shown in dotted lines. The packing is put in to prevent the leakage of steam. Studs *F*, *F*, fitted with nuts and check-nuts, limit the amount of movement of the sliding tube *C*. These studs, or other means of preventing the joint being forced entirely apart by the steam pressure, are extremely important, many very disastrous and fatal accidents having been caused by their absence. The flanges *G* and *H* are bolted to the flanges of the pipe; consequently, when the pipe expands, the body *A* of the expansion joint is forced to the left, and the sliding tube *C* to the

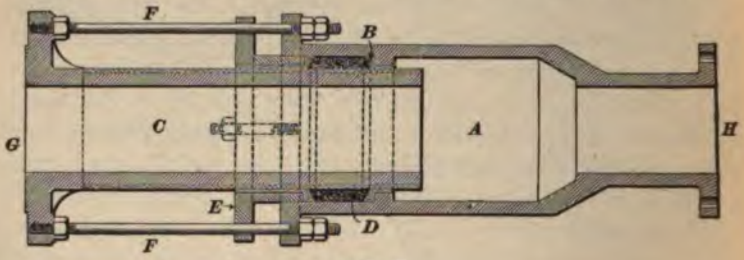


FIG. 29

right, and when the pipe contracts, they are forced in opposite directions. A small drain pipe, not shown in the figure, is usually fitted to the lowest point of the body *A*, and is provided with a stop-cock. This pipe is used to drain the water formed by condensation of the steam from the joint. If no provision is made for carrying off this water, it will soon corrode the body, unless it be made of brass or gun metal, as is sometimes the case.

52. Slip joints are used where the expansion of a straight line of large pipe has to be provided for. In a great many instances, expansion may be provided for by bending the pipes in the manner shown in Fig. 30, the figure illustrating the arrangement of the steam pipes of the steamship Yumuri. There are six Scotch boilers standing athwartship, provided

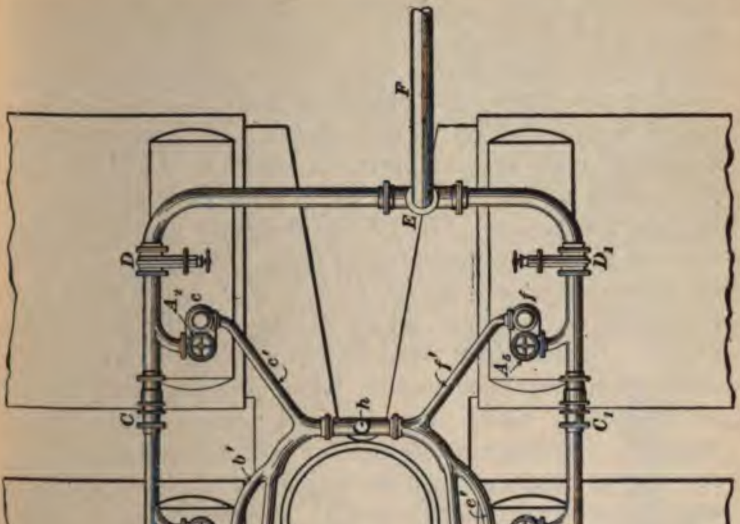


FIG. 30

with steam drums. Attached to the top of the drums are the stop-valves A, A_1, A_2, etc. The steam pipes B, B_1 are bent to the shape shown, and increase in length by merely bending the pipes. Expansion between A_1 and A_2, and A_4 and A_6 is provided for by slip joints C, C_1. Each battery has a separate main steam stop-valve, shown at D and D_1. The two branch mains join at E, whence the main steam pipe F leads to the engine. Fig. 30 also shows the general arrangement of the safety-valve escape pipes. At a, b, c, d, e, and f, the safety valves are shown, a', b', c', d', e', and f' being their respective escape pipes, which deliver into the main escape pipe h. Steam pipes were formerly, but are rarely today, made of copper, each section being provided with flanges, by means of which the different sections were bolted together. Wrought-iron pipes are sometimes employed for the smaller sizes, and, lately, welded-steel pipes have been used, especially in naval vessels. Copper pipes are sometimes strengthened by winding them with one or more layers of square steel wire, their weak spots generally being the brazed joints. Cast iron was at one time very extensively used for steam pipes, but, owing to numerous accidents due to the treacherous nature of the metal, it has been superseded by copper, wrought iron, and steel. Steam pipes are generally covered with some non-heat-conducting material.

MISCELLANEOUS ACCESSORIES

WHISTLES AND SIRENS

53. Every steam vessel is provided with a **whistle** or a **siren** for signaling purposes. Two of the most common constructions are shown in Figs. 31 and 32. Referring to Fig. 31, the bell, or upper portion, is a hollow cylinder closed at the top and open at the bottom, and is held in position by a stud that passes through the center and is secured at the upper end by means of a screw and jam nut. The hollow base has a narrow annular orifice that communicates with

the steam pipe and valve. As the steam rushes out of the orifice in an upward direction, toward the mouth of the bell, it slightly compresses the air contained in the bell. The air being elastic will not retain a fixed or stationary position, but will slightly spring back toward the inrushing steam, when it is again forced back in a compressed state, causing a vibration of the air and steam. These vibrations continue as long as steam is permitted to flow, and are communicated to the surrounding atmosphere, thus producing sound.

The tone may be changed to a higher pitch by lowering, or to a lower pitch by raising, the bell. This may be done by loosening the jam nut and turning the bell up or down, after which the nut should be tightened.

FIG. 31

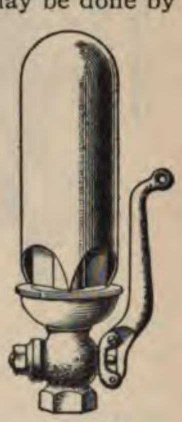


FIG. 32

Whistles are also constructed to produce two or more tones of different pitch simultaneously by dividing the bell into two or more cell-like parts, as shown in Fig. 32. Each apartment produces a different tone, and when these tones chord perfectly, the effect is quite pleasing.

In large steam vessels, the whistle is usually located at a considerable distance above the boilers. In order to prevent the long whistle pipe becoming filled with water, it is advisable to fit a small drain pipe and valve directly above the stop-valve in the whistle pipe, which is placed close to the boiler. When not in use, the steam may be shut off from the whistle, if deemed advisable, and the drain valve opened.

When no separate valve is fitted to the whistle to shut off the steam, the blowing of the whistle, due to an accident to the whistle valve, may be stopped by pushing a stick into the bell; if this is not feasible, due to the construction of the whistle, stuff cotton waste into the bell, using a long stick.

54. The sound produced by a steam siren differs materially from that obtained from a whistle, the latter giving a comparatively shrill and penetrating sound, while that of the former is very deep, far-reaching, and mournful. Steam sirens are often made so that the sound can be projected in any desired direction; such a siren, made by Schaeffer & Budenberg, is illustrated in Fig. 33. The siren consists of a body *a* to which is fitted the movable cowl *b*, which has a gear *c* at its base. A pinion *d* meshes with *c*; the pinion *d* is keyed to a shaft *e* that terminates, in a position convenient to the operator, in a handwheel or crank by which it can be turned, thus turning the cowl. By pulling the cord, which is attached to the lever *f*, a valve inside of *a* is opened and steam admitted to the sound producer. This is a cylindrical drum free to revolve on its axis in suitable bearings. The drum is revolved at high speed by the steam and in doing so produces the sound.

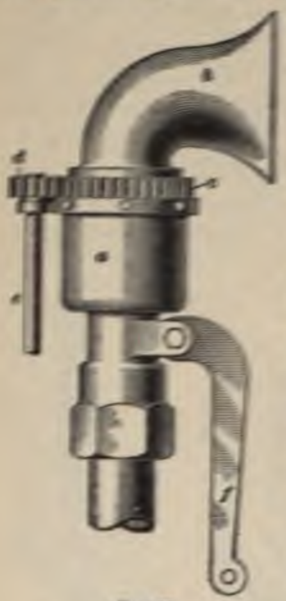


FIG. 33

DONKEY VALVE

55. As the donkey boiler is used only when the vessel is in port, the steam required to work the auxiliary machinery at sea must be taken from the main boiler or boilers, and this is done by connecting one of the main boilers, when several are used, with the main steam pipe leading to the auxiliary machinery. The pipe connecting the boiler with this steam pipe is provided with a stop-valve known as the **donkey valve.** Its construction does not present any special features; in fact, any ordinary globe valve or gate valve may be used.

FIRE-APPARATUS

56. Connected to the main steam pipe or to one of the main boilers is a pipe with branches leading to the different compartments of the vessel, each branch having a separate stop-valve. Usually, all the branches are connected to one fitting, called a *manifold*, as shown in Fig. 34. In case of fire in any of the compartments of the vessel, the compartment is closed, and steam from the boilers is led to the burning compartment by the pipes mentioned, the steam driving out the air and thus smothering the fire.

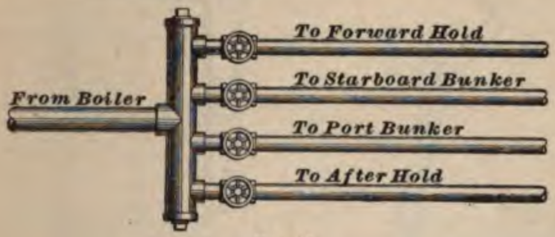


FIG. 34

The rules and regulations of the Board of Supervising Inspectors of Steam Vessels provide that these pipes shall not be less than $1\frac{1}{2}$ inches in diameter, except on steamers employed on western rivers, where the branch pipes must not be less than $\frac{3}{4}$ inch in diameter. Each branch pipe must be supplied with a stop-valve, the handle of which must be marked to indicate the compartment or part of the vessel it leads to, and, if feasible, the whole arrangement is to be enclosed in a suitable box and plainly marked "Fire Apparatus."

FRONT CONNECTION AND SMOKESTACK

57. In Scotch boilers and modifications thereof, the products of combustion, after leaving the furnace, pass through the combustion chamber into the tubes and thence into the front connection. This is made of suitable shape, for instance as shown in Fig. 35, and is built up of sheet

iron about $\frac{3}{16}$ inch thick. It is provided with a large door *A* that is fitted with a baffle plate *B* to prevent radiation. This door gives access to the tubes and front tube sheet. The front connection is usually attached to angle irons *C*, *C* bent to the required shape and secured to the head of the boiler either by studs or by riveting. The front connection, the uptake, and the smokestack are usually made with an air casing around

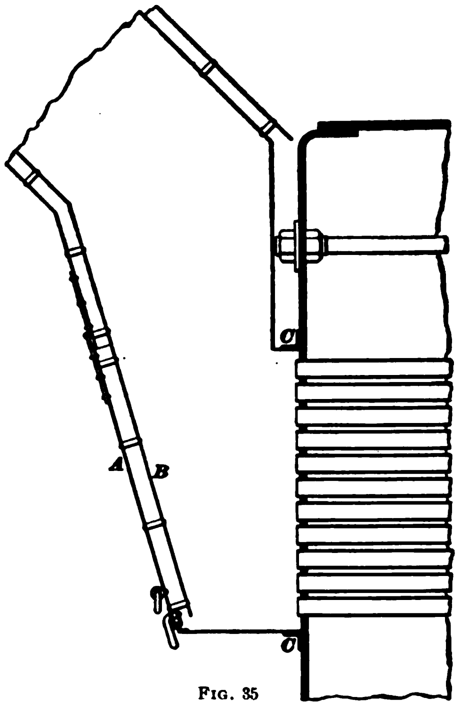


Fig. 35

them; that is, they are made double, leaving an air space of about 3 inches or more between the inner and the outer plates. The lower end of the smokestack is steadied by a cast- or wrought-iron ring secured to the upper deck of the vessel. The top of the smokestack is steadied by guys, usually wire ropes, which may be tightened by means of turnbuckles. The area of the uptake should not be less than the combined area of all the tubes discharging into it.

DAMPER

58. A **damper** is occasionally fitted in the smokestack. The simplest construction of a damper is shown in Fig. 36. A shaft *A* carried in two bearings, one at each side of the stack, has riveted to it a flat wrought-iron plate *B*, elliptic in shape, fitting loosely into the stack. A lever *C* is keyed to one end of the shaft, and provided with an endless chain leading to the engine room, by means of which it may be opened or closed, thus increasing or decreasing the area of

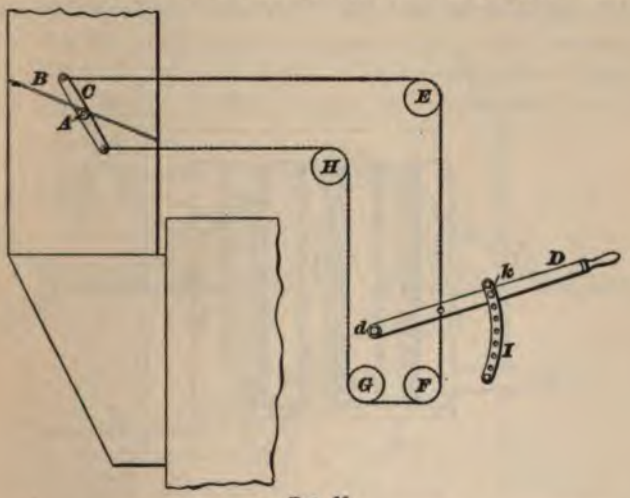


FIG. 36

opening of the stack, and hence regulating the draft. The chain passes over rollers *E*, *F*, *G*, and *H*, and is connected to a lever *D*, working on a stud *d*. A sector *I*, provided with holes, serves to keep the lever in position, a pin *k* being inserted into a hole in the lever and a hole in the sector. With this arrangement, the damper is not affected by the rolling or pitching of the vessel.

SUPERHEATERS

59. Steam may be superheated in a separate vessel, called a **superheater**, utilizing the heat of the waste gases. The use of superheaters in connection with fire-tube boilers was quite common in marine work up to the year 1880, but since then their use has been gradually abandoned, as the practical difficulties incidental to the use of superheated steam are considered to overbalance the advantages. The difficulties encountered are the rapid deterioration of the superheaters, and the carburizing of the lubricant for the engine

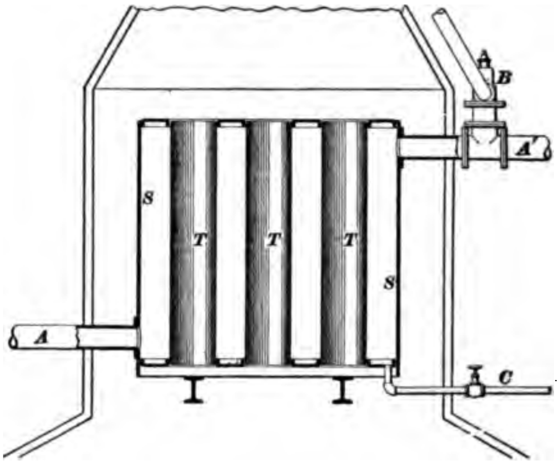


FIG. 37

cylinders. There are many old vessels fitted with superheaters, however, and for this reason two forms are here described.

60. A superheater at one time largely used in British ships is shown in Fig. 37. It consists of a cylindrical shell S having a number of large tubes T passing through it. It is usually fitted into the uptake or the base of the smokestack, and is provided with a safety valve, shown at B, and braced in the same manner as a boiler. It is used for drying the steam on its passage from the boilers to the engine.

This is done by the hot gases of combustion passing through the tubes and around the shell, thus heating the steam in the superheater above the temperature due to its pressure. A drain pipe *C* is provided. The steam enters the superheater through the pipe *A* and leaves it through the pipe *A'*.

The connection to the boiler is made in the following manner: The steam pipes *A*, *A'*, Fig. 38, leading from the several boilers are all joined to the pipe *A''* connected to the bottom of the superheater. The steam enters the superheater *I* through this pipe and passes out at the top in the main steam pipe *G*. Should it be desired to dispense with the use of the superheater, it may be done by opening the valve *C* on the by-pass pipe *D*, and closing the valves *E* and *F*. By means of these valves, the superheated and the

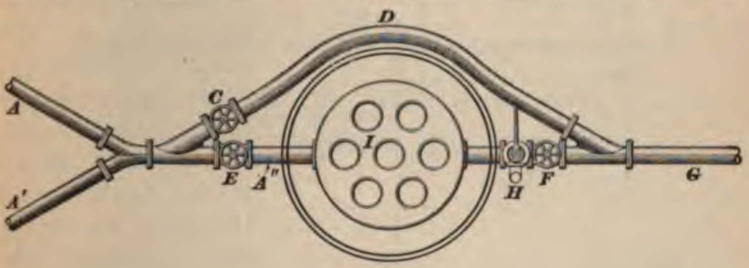


FIG. 38

saturated steam may be mixed, if so desired. This is done by opening the three valves, when part of the steam will pass through the by-pass pipe and part through the superheater, the superheated and saturated steam mixing in the main steam pipe. The superheated steam will have a temperature but little less than that of the gases of combustion, say about 650° F., and this high temperature will soon dry out the packing used about the engine and carbonize the oil used for lubricating the cylinder. To avoid this, the superheated and saturated steam are often mixed in the manner just described, the superheated steam expending part of its heat in drying the saturated steam. The temperature of the mixture will thus be reduced considerably, and drier steam furnished to the engine. The ratio of the heating surface of

the superheater to the heating surface of the boilers is generally made about 1 : 10.

61. Superheaters of the design shown in Fig. 37 have not found much favor in the United States. The **steam chimney**, shown in Fig. 39, is used instead. This, in effect, is nothing but a jacket surrounding the smokestack at its base. The steam enters the steam chimney through the stop-valve *A*, is superheated by coming in contact with the inner lining of the steam chimney, which is heated by the hot gases of combustion, and passes to its destination through the main steam pipe *B*. As usually arranged, the steam chimney cannot be shut off, but all the steam on its passage to the engine must pass through it.

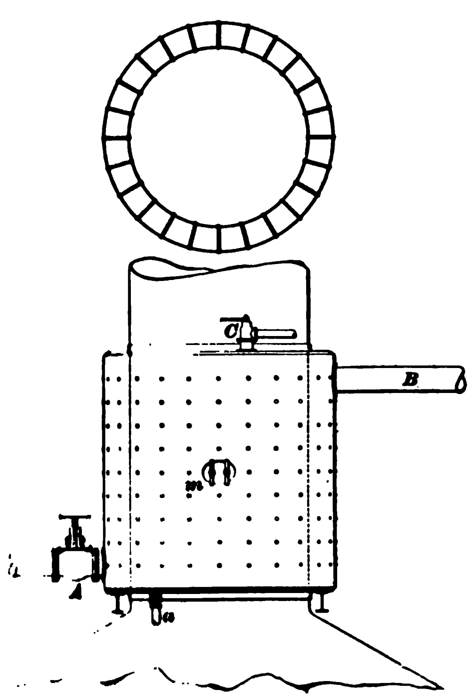


FIG. 39

The steam chimney is braced by stay-bolts of suitable size and pitch, and must be provided with its own safety valve, which is shown at *C*; also, a drain, shown at *a*, by means of which it may be emptied of all the water formed by the condensation of the steam. To allow of inspection and repair, the steam chimney is provided with one or more manholes, according to the size, one of which is shown at *m*.

STEAM DRUM AND DRY PIPE

62. A **steam drum** is a cylindrical vessel connected to the boiler by one or more passages, and placed on top of the boiler to increase the steam space and also to prevent priming. It is supposed that by taking the steam from the boiler at a considerable height above the water level, the steam will be drier. Where steam drums are used, the steam pipe is connected to the drum, and often the safety valves are placed on top of the latter.

Lately, the use of steam drums has been almost abandoned in favor of a so-called **dry pipe**. This is a pipe provided with a number of slots, or perforations, shown at *a*, *a*, etc., Fig. 40. It is placed inside the boiler at the highest point,

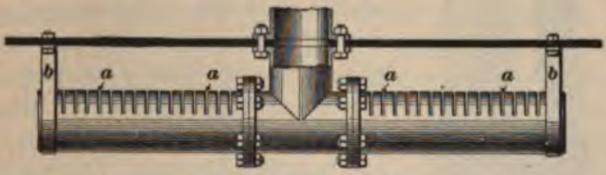


FIG. 40

and is supported at the ends by iron straps *b*, *b* bolted to the boiler shell. The dry pipe is connected to the steam pipe in such a manner that the steam can only enter the steam pipe by passing through the slots, the ends of the pipe being closed. The combined area of the slots, or perforations, in the dry pipe is usually made equal to that of the steam pipe connected to it. A small hole should be drilled into the bottom of the pipe to allow the dry pipe to drain.

STEAM SEPARATORS

63. A **separator** is an apparatus designed to remove the entrained water, or the oil, dirt, or other impurities from a current of steam flowing through a pipe. When the separator is intended simply to free the steam from water, it is placed on the main pipe leading from the boiler to the engine,

and as close as possible to the latter. When it is desired to remove the grease and dirt from the exhaust steam before condensing it and feeding it back into the boiler, the separator is placed in the exhaust pipe leading from the engine to the condenser.

The **Stratton separator** is shown in Fig. 41. It consists of a chamber with a steam inlet and outlet, and containing a vertical pipe *a*. The steam enters by the inlet *c*, and is deflected by a curved partition, which gives it a spiral motion about the pipe *a*. The particles of steam are thrown off by centrifugal action, and run down the walls to the bottom of the chamber. The steam passes through the pipe *a* and out the outlet *d* in a practically dry condition. The separator is provided with a drip pipe *h* for the removal of the water, and a gauge glass *g*. The wings *b*, *b* are four in number, and are for the purpose of destroying the centrifugal effect of the steam after it has reached the bottom of the separator. They likewise offer additional surface for the water particles to adhere to. Were it not for these wings, the steam would keep up its rotative motion while passing up the pipe *a* and thus necessarily carry some of the entrained water with it.

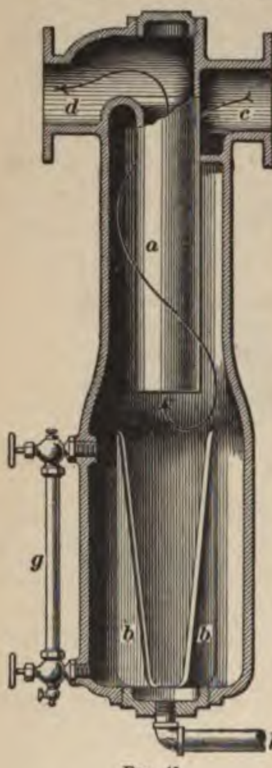


Fig. 41

There are many other makes of separators, all, however, operating on practically the same principle. What is required of a separator is to abruptly change the direction of the current. The particles of water will continue in the original direction of the current by reason of their inertia, while the dry steam passes off in another direction.

BOILER SADDLES

64. The usual method of setting a Scotch boiler when it is placed athwartship is illustrated in Fig. 42 (*a*) and (*b*). Two saddles, or cradles, of wrought-iron or steel plate *a, a* and angle bars *b, b* bent to conform to the shape of the boiler shell are firmly secured to the framing of the vessel by means of the angle bars *b′, b′*. The boiler rests in these saddles and is secured to them, if the boiler is small, by straps passing around it. As the weight of the boiler and the water it contains would throw a heavy bending stress on the saddles when the vessel is rolling, the saddles must be stiffened in an athwartship direction. To do this, gusset braces *c, c* of iron or steel plate are secured by means of the angle-bar clips *d, d* to the saddles and to plates firmly riveted to the reversed frames *e, e*. The diagonal edges of the gusset braces are stiffened by the angle bars *f, f* riveted to them. The lower corners of the gusset-brace plates are cut off to clear the angle bars *b′, b′*. Large boilers are secured to the saddles by means of tap bolts, with the heads on the inside of the boiler. The bolts are screwed into the shell of the boiler from the inside and pass through holes provided for them in the flanges of the saddle angle bars *b, b*; nuts are screwed on the bolts outside the flanges of the angle bars. The holes through the flanges of the angle bars are drilled larger than the bolts, in order to provide room for adjustment. These holes are drilled, after the saddle is permanently secured in its place, from a templet taken from the boiler. Washers are placed under the heads and nuts of these bolts.

It will be observed that the saddle plate is cut away at *g, g*, and that the angle bars are bent to conform to the shape of these spaces, in order to let the saddle clear the butt straps *h, h* that cover the joints in the boiler shell.

65. The setting of a Scotch boiler when it is placed fore and aft is illustrated in Fig. 43 (*a*) and (*b*). The saddles are shown at *a, a*, there being five of them in this case. They are constructed in a similar manner to those described in conjunction with

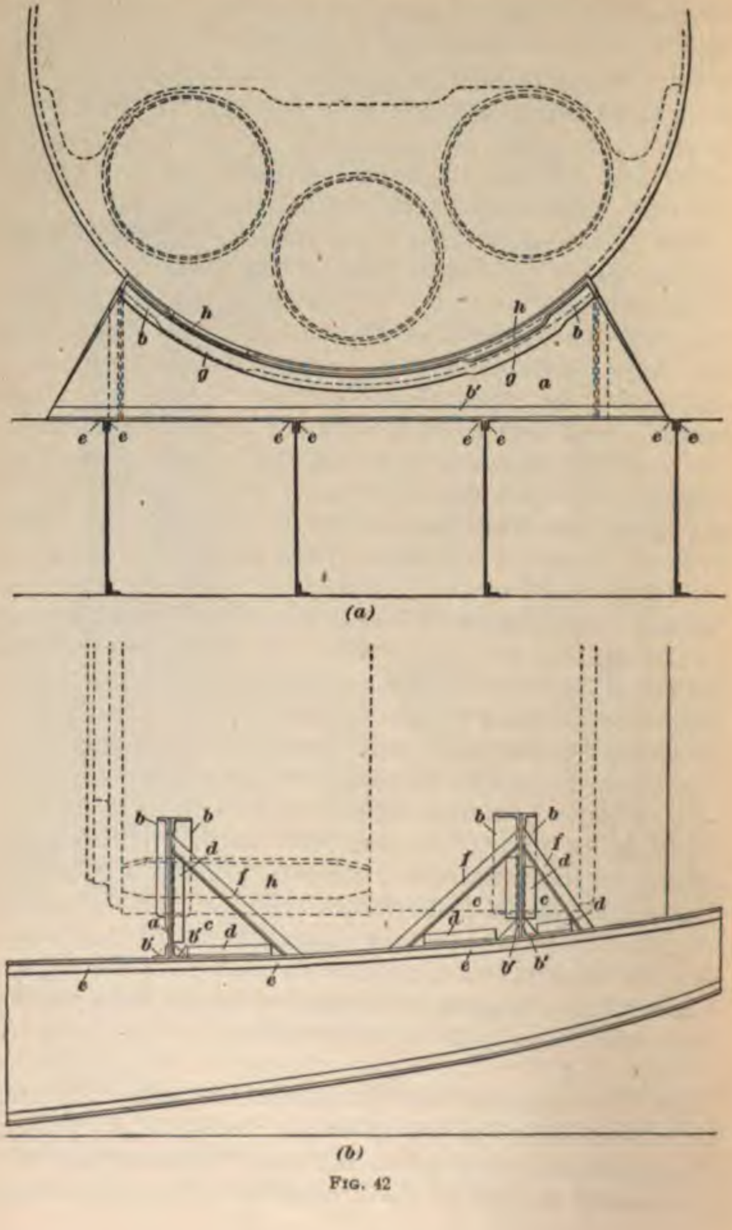
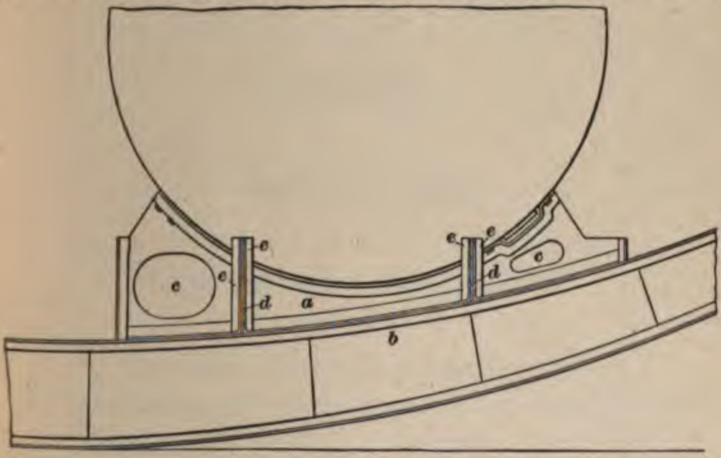
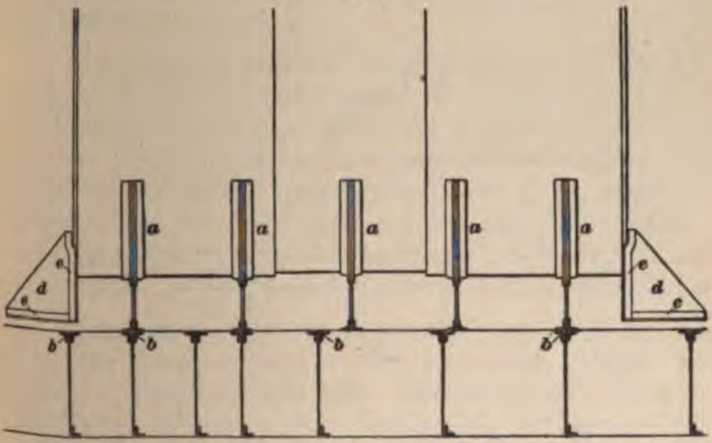


Fig. 42

Fig. 42, with the exceptions that they are shaped to conform to the curve of the reversed frames *b*, *b*, *b*, etc. and that lightening holes *c*, *c* are cut in them to reduce weight. To brace the boiler



(a)



(b)

Fig. 43

when the vessel is pitching, the brackets *d*, *d* are secured to the framing of the vessel and to the ends of the boiler by the angle bars *e*, *e*. The bracket consists of a triangular-shaped piece of iron or steel plate, strengthened by angle bars.

FIRING

COMBUSTION

THEORY OF COMBUSTION

LAWS OF CHEMICAL COMBINATIONS

1. Elements and Compounds.—Every mass of matter is an *element*, a *compound*, or a *mixture*. Iron, silver, sulphur, and oxygen are elements; water, wood, lime, and carbonic acid are compounds.

2. A compound may be decomposed or divided into separate substances. For example, if an electric current is passed through water, the water slowly disappears and two gases are formed. These gases are entirely unlike, and neither resembles the water from which it is produced. Likewise, lime can be divided into two other substances—calcium and oxygen. Any substance that can thus be decomposed or divided into other substances is called a **compound.**

3. There are substances that have never been decomposed into other substances. By no known process can sulphur be separated into other substances; so with iron, gold, arsenic, and many other substances. Substances that have never been decomposed are called **elements.**

The elements that will be considered are: hydrogen, *H*; oxygen, *O*; nitrogen, *N*; carbon, *C*; sulphur, *S*.

In referring to an element, it is customary to use only the symbol, which is usually the first letter of the name. Thus, *H* stands for hydrogen, *C* for carbon, etc.

4. Chemical Combination.—When two or more elements are brought into contact under favorable circumstances, they will combine and form a new substance that is unlike either of the elements. Of course, the new substance will be a compound. Thus, if carbon and oxygen are brought together at a high temperature, they will combine and form carbon dioxide. Hydrogen and oxygen combine to form water. Hydrogen, nitrogen, and oxygen, when combined in certain proportions, form nitric acid. A given volume of nitrogen and three times that volume of hydrogen combine and form ammonia—a gas that differs greatly from both nitrogen and hydrogen.

5. It is supposed that each molecule of an element, such as hydrogen or oxygen, is composed of two atoms. It is further supposed, by chemists, that at a given pressure and temperature equal volumes of all gases, whether simple or compound, contain the same number of molecules. Thus, a cubic foot of hydrogen, a cubic foot of air, a cubic foot of steam, all contain the same number of molecules at the same temperature and pressure.

Suppose, now, that a cubic foot of hydrogen gas is allowed to come into contact with a cubic foot of chlorine gas (symbol, *Cl*). The mixture is exposed to heat or light, and the gases combine. The process of combination is explained as follows: There is a certain attraction or affinity between the hydrogen atoms and the chlorine atoms. Under the influence of heat or light, this attraction becomes so strong that the two atoms composing the molecule of hydrogen are torn apart. Likewise, the atoms composing a molecule of chlorine separate. Each atom of chlorine seizes on an atom of hydrogen and forms a molecule of an entirely new gas, viz., hydrochloric-acid gas. Since each atom of chlorine takes *one* atom of hydrogen, it is plain that the number of molecules of each gas must be the same. In other words, 1 cubic foot of chlorine requires 1 cubic foot of hydrogen to combine with it; these gases cannot be made to combine in any other proportion. For example, if 3 cubic feet of

chlorine were placed in contact with 2 cubic feet of hydrogen, 4 cubic feet of hydrochloric-acid gas would be formed, and the extra cubic foot of chlorine would still remain chlorine. The symbol for hydrochloric-acid gas is HCl.

Suppose, now, that hydrogen and oxygen are placed in contact and heated. They will combine and form steam (or water); but it will be found that each atom of oxygen seizes two atoms of hydrogen to form a molecule of water, and therefore the volume of hydrogen must be double the volume of the oxygen with which it combines. This is shown by the symbol for water, which is H_2O; that is, a molecule of water is composed of two atoms of hydrogen to one of oxygen. Similarly, the symbol for ammonia is NH_3; that is, three atoms of hydrogen to one of nitrogen. Again, hydrogen and carbon form a compound; each atom of carbon seizes four atoms of hydrogen and forms a molecule of marsh gas. The symbol for marsh gas is, therefore, CH_4.

6. The symbol of any compound indicates how the atoms of the elements combine to form the compound. Thus, the symbol for water, H_2O, shows that two atoms of hydrogen and one of oxygen unite to form a molecule of water. The symbol H_2SO_4 (sulphuric acid) shows that one molecule of the sulphuric acid contains two atoms of hydrogen, one of sulphur, and four of oxygen.

7. Combination by Weight.—One cubic foot of hydrogen combines with just 1 cubic foot of chlorine. But on weighing each gas it is found that the cubic foot of chlorine weighs 35.5 times as much as the cubic foot of hydrogen. A cubic foot of oxygen weighs 16 times as much as a cubic foot of hydrogen.

At a given pressure and temperature, equal volumes of gases contain the same number of molecules; therefore, 1 cubic foot of oxygen must contain the same number of atoms as 1 cubic foot of hydrogen. Now, since the former weighs 16 times as much as the latter, it follows that an atom of oxygen weighs 16 times as much as an atom of hydrogen. Similarly, an atom of chlorine weighs 35.5 times

as much as an atom of hydrogen. This ratio between the weight of an atom of any element and the weight of an atom of hydrogen is called the **atomic weight** of the element. The atomic weight of any element may be found by dividing the weight of a given volume, say 1 cubic foot of the element when in a gaseous state, by the weight of 1 cubic foot of hydrogen when both are at the same temperature and pressure. The atomic weight is, therefore, much the same thing as specific gravity, except that the weight of hydrogen is used as the standard of comparison instead of the weight of water.

8. The atomic weights of the elements named are: Hydrogen, H, 1; oxygen, O, 16; nitrogen, N, 14; carbon, C, 12; sulphur, S, 32.

By the aid of these atomic weights, the composition of any substance, by weight, known as the **molecular weight,** can be found when its symbol is known. The molecular weight may be defined as the ratio of the weight of a molecule of a substance to the weight of an atom of hydrogen, the weight of the latter being taken as 1. Take water, symbol H_2O; that is, there are two atoms of H to one of O. Multiply the number of atoms of each by the atomic weight of the atom. Thus,

$$\begin{aligned} 2 \times 1 &= 2 \text{ parts by weight of hydrogen} \\ 1 \times 16 &= \underline{16} \text{ parts by weight of oxygen} \\ & 18 \text{ parts by weight of water} \end{aligned}$$

Hence, the water is composed of $\frac{2}{18} = .1111 = 11.11$ per cent. of hydrogen and $\frac{16}{18} = .8889 = 88.89$ per cent. of oxygen.

As another example, take carbon dioxide, CO_2. Then,

$$\begin{aligned} &1 \text{ atom of } C \times \text{atomic weight, } 12 = 12 \text{ parts by weight of } C \\ &2 \text{ atoms of } O \times \text{atomic weight, } 16 = \underline{32} \text{ parts by weight of } O \\ &\phantom{2 \text{ atoms of } O \times \text{atomic weight, } 16 =} 44 \text{ parts by weight of } CO_2 \end{aligned}$$

Hence, CO_2 contains $\frac{12}{44} = .2727 = 27.27$ per cent. carbon, and $\frac{32}{44} = .7273 = 72.73$ per cent. oxygen. From these

examples, it is plain that the molecular weight of water is 18 and of carbon dioxide 44.

9. Mixtures.—Two or more substances, either elements or compounds, may be mixed together and yet not combine to form a new substance. They are then said to form a **mixture.** The mixture has the properties of the substances composing it. The most familiar example of a mixture is ordinary air. It is composed of oxygen and nitrogen, 23 parts, by weight, of the former to 77 parts, by weight, of the latter. The two gases are not combined chemically; they are simply mixed.

ELEMENTS OF COMBUSTION

10. Definitions.—**Combustion** is a very rapid chemical combination. The atoms of some of the elements have a very great affinity or attraction for those of other elements, and when they combine they rush together with such rapidity and force that heat and light are produced. Oxygen, for example, has a great attraction for nearly all the other elements. An atom of oxygen is ready to combine with almost any substance with which it comes in contact. For carbon, oxygen has a particular liking, and whenever these two elements come in contact at a sufficiently high temperature, they combine with great rapidity. The combustion of coal in the furnace of a boiler is of this nature. The temperature of the furnace is raised by kindling the fire, and then the carbon of the coal begins to combine with oxygen taken from the air. The combination is so rapid and violent that a great quantity of heat is given out.

The elements that enter into combustion are oxygen and, usually, either carbon or hydrogen. Coal, wood, and other fuels are composed almost entirely of these three elements. Combustion is, therefore, a rapid chemical combination of oxygen with either carbon or hydrogen, or both.

11. When carbon and oxygen combine they form CO_2, or carbon dioxide; when hydrogen and oxygen combine they form water, H_2O. These are called the **products of**

combustion. When, as is ordinarily the case, the oxygen is obtained from the air, the nitrogen of the air passes into the furnace with the oxygen. It takes no part in the combustion, but passes through the furnace and up the smokestack with the CO_2 without any change in its nature; it is, however, usually called a *product of combustion in air.*

12. Weight and Volume of Air Required for Combustion.—Carbon dioxide, CO_2, is composed by weight of 12 parts of carbon and 32 parts of oxygen. Hence, to burn a pound of carbon requires $\frac{32}{12} = 2\frac{2}{3}$ pounds of oxygen. If the oxygen is taken from the air, it will take $2\frac{2}{3} \div .23 = 11.6$ pounds of air to supply the $2\frac{2}{3}$ pounds of oxygen. This is because only 23 per cent. of air is oxygen. The combustion of a pound of carbon may be represented as follows:

	Elements	Products
1.0 pound carbon .	1.00 pound carbon .	3.67 pounds CO_2
11.6 pounds air . . .	2.67 pounds oxygen .	
	8.93 pounds nitrogen	8.93 pounds nitrogen
12.6	12.60	12.60

That is, 1 pound of carbon requires 11.6 pounds of air for complete combustion. Of this air, 2.67 pounds is oxygen, which combines with the pound of carbon, forming 3.67 pounds of carbon dioxide. The 8.93 pounds of nitrogen contained in the air passes off with the CO_2 as a product of combustion.

Take, next, the complete combustion of 1 pound of hydrogen. The product of the combustion is water, H_2O, which is composed, by weight, of 2 parts hydrogen to 16 parts oxygen. Hence, 1 pound of H requires $\frac{16}{2} = 8$ pounds of O to unite with it. The air required to furnish 8 pounds of O is $8 \div .23 = 34.8$ pounds. The process of combustion is, therefore, as follows:

	Elements	Products
1 pound hydrogen	1 pound hydrogen	9 pounds water (H_2O)
34.8 pounds air . . .	8 pounds oxygen	
	26.8 pounds nitrogen	26.8 pounds nitrogen
35.8	35.8	35.8

13. There is one other case that may occur; the combustion of carbon may not be complete. If insufficient air or oxygen is supplied to the burning carbon, it is possible for the carbon and oxygen to form another gas, carbon monoxide, CO, instead of carbon dioxide, CO_2.

The combustion of 1 pound of carbon to form CO, of course, requires only one-half the oxygen that would be necessary to form CO_2. This is because in CO gas one atom of carbon seizes one atom of oxygen instead of two atoms. To burn 1 pound of carbon to CO_2 requires 11.6 pounds of air; to burn it to CO will, therefore, require but 5.8 pounds of air.

14. The quantities of air required for combustion are shown in the following scheme:

1 Pound	Air at 62°	Product of Combustion
Hydrogen . .	34.8 pounds, or 457 cubic feet	Water Nitrogen
Carbon burned to CO_2 . . .	11.6 pounds, or 152 cubic feet	Carbon dioxide Nitrogen
Carbon burned to CO . . .	5.8 pounds, or 76 cubic feet	Carbon monoxide Nitrogen

15. The fuels in common use are composed chiefly of carbon, with sometimes a small percentage of hydrogen, oxygen, and incombustible matter called *ash*. When the percentages of carbon and hydrogen are known, the air required for the combustion of 1 pound of the fuel is easily found. For example, suppose that a certain coal is 91 per cent. carbon and 9 per cent. hydrogen. To burn the carbon requires $152 \times .91 = 138.32$ cubic feet of air; to burn the hydrogen requires $457 \times .09 = 41.13$ cubic feet of air. Hence, to burn 1 pound of the fuel requires $138.32 + 41.13 = 179.45$ cubic feet of air.

From this the following rule is derived:

Rule.—*To find, in cubic feet, the quantity of air at 62° F. required to burn 1 pound of a given fuel, multiply the percentage of carbon by 152, and the percentage of hydrogen by 457. Add the two products.*

Or, $A = 152\,C + 457\,H$

where A = air, in cubic feet;

C = percentage of carbon, expressed decimally;

H = percentage of hydrogen, expressed decimally.

EXAMPLE 1.—How many cubic feet of air are required to burn 1 pound of coal containing 84 per cent. carbon, 5 per cent. hydrogen, 7 per cent. oxygen, and 4 per cent. ash?

SOLUTION.—Applying the rule,

$A = 152 \times .84 + 457 \times .05 = 150.53$ cu. ft. Ans.

EXAMPLE 2.—How many cubic feet of air are required to burn 1 pound of coal oil containing 88 per cent. carbon, 11 per cent. hydrogen, and 1 per cent. oxygen?

SOLUTION.—Applying the rule,

$A = 152 \times .88 + 457 \times .11 = 184.03$ cu. ft. Ans.

When the fuel already contains oxygen, a little less air than given by the rule is required to burn it; if it contains sulphur, a little more air will be required. In either case, the difference is very slight. It will be found that 1 pound of coal requires practically the same amount of air, whether it be anthracite or bituminous. Roughly speaking, it requires about 12 pounds, or 160 cubic feet, of air to burn 1 pound of carbon or coal. If less air is supplied, the combustion is imperfect; that is, the carbon burns to CO instead of CO_2.

16. Heat of Combustion.—The quantity of heat developed by the complete combustion of a pound of fuel is known as its **heat of combustion**, and also as its *heating value*, *heating power*, *calorific power*, or *calorific value*. The quantities of heat produced by the complete combustion of the elements composing the fuels have been found by experiment. They are: Hydrogen, 62,000 British thermal units per pound; carbon burned to CO_2, 14,600 British thermal units per pound; carbon burned to CO, 4,400 British thermal units per pound; sulphur, 4,000 British thermal units per pound. When a fuel contains oxygen, the oxygen during combustion will unite with one-eighth its weight of hydrogen and form water, H_2O, thus reducing the heat of combustion of the hydrogen. Suppose a fuel contains, by weight, 85 per cent. carbon, 4 per cent. oxygen, 6 per cent. hydrogen, 1 per

cent. sulphur, and 4 per cent. ash. The total heat of combustion of a pound of this fuel is found thus: The heat of combustion of the carbon is $14,600 \times .85 = 12,410$ British thermal units. The heat of combustion of the hydrogen, remembering that the oxygen present combines with one-eighth of its weight of hydrogen, is $62,000 \times \left(.06 - \frac{.04}{8}\right)$ $= 3,410$ British thermal units. The heat of combustion of the sulphur is $4,000 \times .01 = 40$ British thermal units. Then, the total heat of combustion is $12,410 + 3,410 + 40 = 15,860$ British thermal units. Expressing this in the form of a rule, **Dulong's rule** is obtained, which is as follows:

Rule.—*To find the heat of combustion of a pound of a given fuel, multiply 14,600 by the percentage of carbon; divide the percentage of oxygen by 8, subtract the quotient from the percentage of hydrogen, and multiply 62,000 by the remainder; multiply 4,000 by the percentage of sulphur, and add the three products.*

Or, $$X = 14,600\ C + 62,000 \left(H - \frac{O}{8}\right) + 4,000\ S$$

where X = heat of combustion per pound, in British thermal units;
C = percentage of carbon, expressed decimally;
H = percentage of hydrogen, expressed decimally;
O = percentage of oxygen, expressed decimally;
S = percentage of sulphur, expressed decimally.

EXAMPLE.—What is the heat of combustion of a pound of fuel containing 66 per cent. carbon, 8 per cent. oxygen, 8 per cent. hydrogen, 2 per cent. sulphur, and 16 per cent. ash?

SOLUTION.—Applying the rule,

$$X = 14,600 \times .66 + 62,000 \left(.08 - \frac{.08}{8}\right) + 4,000 \times .02$$
$$= 14,056 \text{ B. T. U.} \quad \text{Ans.}$$

17. Maximum Evaporation.—It requires 966.1 British thermal units to evaporate 1 pound of water at 212° F. into steam of the same temperature and corresponding pressure. Then, the greatest weight of water, that is, the theoretical weight, that can be evaporated from and at 212° F. by a

pound of a given fuel is found by dividing its heat of combustion per pound by 966.1.

EXAMPLE.—How many pounds of water can be evaporated, theoretically, by a pound of coal whose heat of combustion is 13,897 British thermal units?

SOLUTION.—Evaporation $= \frac{13,897}{966.1} = 14.38$ lb. Ans.

18. Temperature of Combustion.—Making no allowance for losses of heat, and supposing that just enough air is furnished for the combustion, burning carbon should have a temperature of about 4,940° F. above zero; burning hydrogen should have a temperature of about 5,800° F. above zero. In practice, these temperatures are never attained, on account of the losses of heat. Usually, the quantity of air admitted to the furnace is from 50 to 100 per cent. more than is theoretically necessary for the combustion. This extra quantity of air enters at a temperature of 60° F. or 70° F., and escapes up the smokestack at a temperature of from 400° F. to 600° F. A large quantity of heat is thus wasted and the temperature of the fire is greatly lowered. When the fire is outside the boiler and the furnace is surrounded by brickwork, the furnace temperature may be 2,500° F. or 3,000° F., but when the furnace is inside the boiler and is surrounded on all sides by water, the temperature rarely rises above 2,000° F., and is usually less. A high temperature is desirable, since the water of the boiler will take up heat much faster at high furnace temperatures than at low furnace temperatures; combustion is also more perfect at high temperatures.

EXAMPLES FOR PRACTICE

1. How many pounds of air will be required for the perfect combustion of 7 pounds of carbon? Ans. 81.2 lb.

2. A fuel is 88 per cent. carbon and 12 per cent. hydrogen; how many cubic feet of air are required for the complete combustion of 1 pound of the fuel? Ans. 188.6 cu. ft.

3. (*a*) How many British thermal units would the combustion of the pound of fuel of example 2 give out? (*b*) How many pounds of water at 212° F. would 1 pound of this fuel evaporate?

Ans. { (*a*) 20,288 B. T. U.
(*b*) 21 lb.

(4) The chemical symbol of the product of combustion of sulphur with oxygen is SO_2 (sulphurous oxide); what is the composition of this gas by weight?

Ans. $\begin{cases}\text{Sulphur, 50 per cent.}\\ \text{Oxygen, 50 per cent.}\end{cases}$

(5) Assume that, with ordinary draft, double the theoretical quantity of air is used to burn a fuel; how many cubic feet of air will be required to burn 115 pounds of coal, the chemical composition being H, 5 parts; C, 90 parts; O, 3 parts; and ash, 2 parts; total, 100 parts.

Ans. 36,719.5 cu. ft.

6. What is the heat of combustion of a pound of coal having the composition mentioned in example 5? Ans. 16,007.5 B. T. U.

FUELS AND THEIR COMBUSTION

KINDS OF FUELS

19. Coal.—The **fuels** ordinarily used in marine work in the generation of steam are coal, wood, and oil. A prominent authority, Mr. William Kent, divides coal into four leading varieties, as follows:

1. *Anthracite*, which contains from 92.31 to 100 per cent. of fixed carbon and from 0 to 7.69 per cent. of volatile hydrocarbons.

2. *Semianthracite*, which contains from 87.5 to 92.31 per cent. of fixed carbon and from 7.69 to 12.5 per cent. of volatile hydrocarbons.

3. *Semibituminous coal*, which contains from 75 to 87.5 per cent. of fixed carbon and from 12.5 to 25 per cent. of volatile hydrocarbons.

4. *Bituminous coal*, which contains from 0 to 75 per cent. of fixed carbon and from 25 to 100 per cent. of volatile hydrocarbons.

20. Anthracite is rather hard to ignite and requires a strong draft to burn it. This coal is quite hard and shiny. In color, it is a grayish black, and burns with almost no smoke; this fact gives it a peculiar value in places where smoke is objectionable.

Anthracite is known to the trade by different names, according to the size into which the lumps are broken.

These names, with the generally accepted dimensions of the screens over and through which the lumps of coal will pass, are:

Culm passes through $\frac{3}{16}$-inch round mesh.

Rice passes over $\frac{3}{16}$-inch mesh and through $\frac{3}{8}$-inch square mesh.

Buckwheat passes over $\frac{3}{8}$-inch mesh and through $\frac{1}{2}$-inch square mesh.

Pea passes over $\frac{1}{2}$-inch mesh and through $\frac{3}{4}$-inch square mesh.

Chestnut passes over $\frac{3}{4}$-inch mesh and through $1\frac{3}{8}$-inch square mesh.

Stove passes over $1\frac{3}{8}$-inch mesh and through 2-inch square mesh.

Egg passes over 2-inch mesh and through $2\frac{3}{4}$-inch square mesh.

Broken passes over $2\frac{3}{4}$-inch mesh and through $3\frac{1}{2}$-inch square mesh.

Steamboat passes over $3\frac{1}{2}$-inch mesh and out of screen.

Lump passes over bars set from $3\frac{1}{2}$ to 5 inches apart.

21. Semianthracite kindles easily and burns more freely than the true anthracite; hence, it is highly esteemed as a fuel. It crumbles readily, and may be distinguished from anthracite by the fact that when just fractured it will soil the hand, while anthracite will not. It burns with very little smoke. Semianthracite is broken into different sizes for the market; these sizes are the same, and are known by the same trade names, as the corresponding sizes of anthracite.

22. Semibituminous coal differs from semianthracite only in having a smaller percentage of fixed carbon and more volatile hydrocarbons. Its physical properties are practically the same, and since it burns without the smoke and soot emitted by bituminous coal, it is a valuable steam fuel.

23. Bituminous coal may be broadly divided into three general classes:

1. *Caking Coal*.—This name is given to coals that, when burned in the furnace, swell and fuse together, forming a

spongy mass that may cover the whole surface of the grate. These coals are difficult to burn, since the fusing prevents the air passing freely through the bed of burning fuel; when caking coals are burned, the spongy mass must be frequently broken up with the slice bar, in order to admit the air needed for its combustion.

2. *Free-Burning Coal.*—This is often called *non-caking coal*, from the fact that it has no tendency to fuse together when burned in a furnace.

3. *Cannel Coal.*—This is a grade of bituminous coal that is very rich in hydrocarbons. The large percentage of volatile matter makes it valuable for gas making, but it is little used for the generation of steam, except near the places where it is mined.

Bituminous and semibituminous coals are known to the trade by the following names:

Lump, which includes all coal passing over screen bars $1\frac{1}{2}$ inches apart.

Nut, which passes over bars $\frac{3}{4}$ inch apart and through bars $1\frac{1}{2}$ inches apart.

Pea, which passes over bars $\frac{3}{8}$ inch apart and through bars $\frac{3}{4}$ inch apart.

Slack, which includes all coal passing through bars $\frac{3}{8}$ inch apart.

24. Lignite may be classified as coming under the general head of bituminous coal. Properly speaking, lignite occupies a position between peat and bituminous coal, being probably of a later origin than the latter. It has an uneven fracture and a dull luster. The value of lignite as a steam fuel is limited, since it will easily break in transportation. Exposure to the weather causes lignite to absorb moisture rapidly, when it will crumble quite readily. Lignite is non-caking and yields but a moderate heat, and is in this respect inferior to even the poorer grades of bituminous coal.

25. The heat of combustion of coal depends entirely on its chemical composition, and as this varies between wide limits, the heat of combustion also varies between

corresponding limits. Thus, a sample of Arkansas lignite had a value of 9,215 British thermal units per pound; a sample of Kentucky cannel coal, 15,198 British thermal units per pound; a sample of Pennsylvania coal from Monongahela, 14,130 British thermal units per pound.

26. Wood.—Steamers navigating rivers flowing through localities where wood is abundant and coal either very scarce, unobtainable, or very high priced, often use wood for steam making. The heating value of the different woods varies but little when dried; 1 pound of wood may be estimated to be equal in steam-making capacity to about .4 pound of ordinary coal. Naturally the heating value of wood varies considerably with its condition; thus, when full of sap, as when a live tree has just been cut up, its heating value is much less than when the largest part of the moisture has been driven off by seasoning the wood or artificially drying it.

27. Mineral Oil.—The mineral oil known as petroleum is occasionally used in marine work as a fuel, and as such has many advantages over coal. The universal use of oil, however, is prevented chiefly by the limited supply, and to some extent by its high price, except under especially favorable conditions. The advantages of oil as a fuel in marine work are as follows: Reduced weight and space per horsepower; decreased number of firemen; reduction in time required for raising steam; instantaneous lighting and extinguishment of the fire; ready adjustment of the fire to suit the demand for steam; absence of ashes and smoke. The disadvantages are: Loss of fuel by evaporation; danger of explosion; unpleasant odors; difficulty of obtaining a supply everywhere; comparatively high price.

The chemical composition of oil, like that of coal, varies through a considerable range, but the following may be considered as average: carbon, 84 to 88 per cent.; hydrogen, 11 to 14 per cent.; and oxygen, .1 to 1.5 per cent. The specific gravity at 32° F. is approximately .9, and the average calorific value, or heating power, may be taken at 20,000 British thermal units per pound of oil. If it is

assumed that good coal on an average will develop 14,000 British thermal units per pound, it is plain that 1 pound of oil is equivalent to $\frac{20,000}{14,000} = 1.4$ pounds, nearly, of coal in heating value. There are features of oil burning that modify the above relative value and often turn the balance in favor of the use of coal, but so far as the actual quantity of heat produced is concerned, it may be safely stated that oil yields about 40 per cent. more than an equal weight of coal.

COMBUSTION OF COAL

28. Systems of Firing.—The management of the fire or fires used in generating steam is known as **firing**. The style of firing to be adopted in any given case depends largely on the conditions present, such as the kind of fuel used, the intensity of the draft, the demand for steam, etc.

There are three methods of hand firing, known as *coking firing*, *spreading firing*, and *alternate firing*, in common use. Each of these methods has advantages peculiar to itself, and none is applicable to all cases and all conditions, that is, if economy in the generation of steam is an object.

29. The **coking system** of firing is especially adapted to bituminous coals that are rich in volatile matter. The coal is first piled on the dead plate near the door and there allowed to coke. After coking from 20 to 30 minutes, the hydrocarbons, that is, the volatile constituents of the coal that can be distilled or driven off by heat, will have been driven off. The coke is then pushed toward the bridge and distributed evenly over the fire. A new charge of coal is immediately heaped on the dead plate.

This is one of the most economical methods of burning bituminous coal; if properly managed, it will give excellent results in regard to the prevention of smoke. In order to get good results, the furnace door should be perforated and a suitable damper fitted for opening and closing the perforations. The air admitted in jets through the openings mixes intimately with the gases formed; the mixture passes

to the rear over the bed of burning coke on the grate, where the temperature is high enough to secure their ignition and complete the combustion before they are chilled by contact with the cold surfaces of the boiler and tubes. To secure success with this method, the coal should be charged in small quantities and allowed to remain on the dead plate until it is as thoroughly coked as possible; 30 minutes will, in general, be sufficient. As a matter of course, actual trial in each and every case will have to determine the proper length of time. Large lumps that will coke slowly must be broken up; if the coal cakes badly in coking, the crust thus formed must be broken with the slice bar from time to time, so as to secure the complete removal of the hydrocarbons. The size of the grate and the intensity of the draft should be such that the coke will be burned at as high a rate of combustion, per square foot of grate surface, as the conditions will permit. This results in a high furnace temperature, which promotes complete combustion of the gases.

Coking firing is best adapted for cases where the demand for steam is moderately regular, since with coking firing it is somewhat difficult to force the boiler when there is a sudden and heavy demand for steam. Coking firing should never be adopted for anthracite.

30. The **spreading system** of firing consists of covering the whole of the grate evenly with the fresh charge of coal, and is the system in most common use. While good results can be obtained by it, if the firing is done skilfully, the spreading system is not particularly to be recommended either for economical or for smokeless combustion. Best results will be obtained from the spreading system by firing light charges at frequent intervals. The habit of covering the incandescent coke on the grate with a thick layer of fresh coal naturally results in a lowering of the furnace temperature far below the ignition point of the hydrocarbons driven off. In consequence, there is an enormous waste of heat, and with bituminous coal, vast quantities of black smoke are produced. To prevent this heat loss, the firing must be light and frequent.

The spreading system is best adapted to anthracite in sizes larger than pea.

31. In the **alternate system** of firing, the coal is thrown alternately on each side of the furnace; at one firing one side of the grate is spread with coal, and at the next firing the other side receives the charge. This method is preferable to the spreading system in that the whole of the furnace is not cooled off at once by the fresh fuel. While it keeps a bright bed of fuel in one side of the furnace and tends to keep the average temperature of the furnace nearly constant, it cannot be recommended as being the best method for securing complete combustion of the hydrocarbons that form a valuable constituent of bituminous coal. The gases from the freshly fired coal, instead of being passed over the bright bed of fuel on the other side of the furnace, are likely to pass directly to the smokestack without being sufficiently heated to secure their ignition and complete combustion. For both bituminous and anthracite coals, the alternate system of firing is preferable to the spreading system, however, since gas explosions in the furnace are not as likely to occur as when the latter system is used.

32. Gas Explosions.—Explosions of the gases in the furnace, commonly called *back draft*, occur usually with small coal and are the result of careless firing. When the smaller sizes of anthracite or bituminous coal are burned with the spreading system, and when a heavy charge is put into the furnace, it frequently happens that an explosive mixture of air and gas is formed, which needs but a spark to ignite it. Owing to the interstices between the pieces of coal being small and tortuous, especially with the smaller coals, the hydrocarbons driven off from the heavy charge are not ignited as rapidly as formed, and hence collect and mix with the air above the grate, forming an explosive mixture if the conditions are favorable. All danger of a gas explosion is obviated if the firing is done very lightly, or if the alternate system is adopted, or if some part of the fire is left uncovered when putting in fresh coal, thus igniting the hydrocarbons as

quickly as they are distilled off. The smaller the size of the coal, the greater is the liability of a gas explosion, with a heavy charge fired spreading. With coals of sizes larger than pea, there is little danger of an explosion when fired spreading, except when fired thick instead of light.

33. Thickness of Fire.—The thickness of the fire in a furnace depends to a large extent on the draft, the nature and the quality of the fuel, the size of the grate, and the rate of combustion required. As a general rule, the stronger the draft, the thicker the fire can be. Where a high rate of combustion is required with ordinary natural draft, it may often be attained by a thin fire and frequent, light charges of fresh coal. It is claimed that this is not the most economical way of firing, because the cold air rushing into the furnace whenever the fire-door is open for charging the fire reduces the temperature of the furnace and of the gases in combustion, and consequently reduces the evaporation. For instance, let 10 pounds of coal be burned per square foot of grate surface per hour, evaporating 100 pounds of water. Then, if 20 pounds of coal be burned on the same area in the same time, instead of evaporating twice the amount of water, that is, 200 pounds, the actual evaporation will probably be about 170 or 180 pounds. The average thickness of the fire varies from 8 to 14 inches.

34. Fire-Tools.—The special tools used in working the fires are shown in Fig. 1. The **slice bar,** shown at *A*, is made of 1-inch or 1⅛-inch round iron, about 8 feet long, one end of it being flattened and a handle formed at the other end. This tool is employed for breaking up the crust formed at the surface of the fire when bituminous coal is used. The heat of the fire fuses the fresh charge of coal in a short time, thus keeping a sufficient supply of air from passing through the grate. About ten minutes after charging the fire, the slice bar is run into the fire on the surface of the grate, and, by depressing the end of the bar outside the furnace, the fire is broken up. This operation is called *slicing the fire.*

The **T bar**, shown at *B*, is a slice bar with a broad, flat point, and is used, when firing anthracite, for breaking up the clinkers and cinders lying on the grate. It is merely run along the top of the grate the whole length, this operation facilitating the admission of air to the fire.

The **hoe**, made of $\frac{7}{8}$-inch bar iron, is shown at *C*; generally two are used—a heavy hoe with a broad end, and a lighter one. The lighter one is used for leveling the fire; the heavier one in cleaning the fire and ash-pit.

The usual construction of the **poker** is shown at *D*. A handle is formed at one end; the other end is enlarged and is rectangular in cross-section. It is provided with a slot

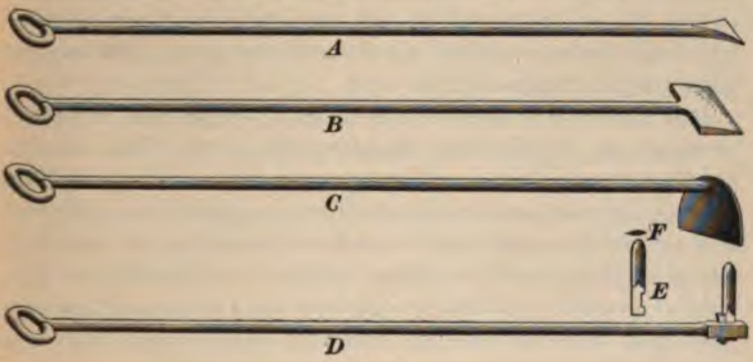


FIG. 1

for the insertion of the blade, which is shown enlarged at *E*. These blades are usually $\frac{5}{16}$ inch thick by $1\frac{1}{4}$ inches wide, and about 8 inches long, and are forged to the shape shown at *F*. A number of these blades are carried in stock, and are inserted when required by driving out the taper key, thus allowing the old blade to be removed and a new one inserted; the key is then driven home, locking the blade in position. This tool is used for cleaning the space between the grate bars from below, should it become choked.

In some localities, a so-called **devil's claw** is added to the equipment. This is simply a three-pronged rake used for drawing large clinkers out of the fire, but as this can

be done with a hoe just about as well, the devil's claw has not come into very extensive use.

35. Cleaning Fires.—Two methods of cleaning a fire are in use. In the first method, the burning fuel, or the *live coal*, as it is termed, is pushed to one side of the furnace while the other side is being cleaned. The live coal is then distributed evenly all over the grate and covered with a light charge of fresh fuel.

In the second method, the live coal is pushed back against the bridge, and the cinders and clinkers covering the front of the grate are pulled out. The live coal is then pulled to the front of the furnace, and the cinders, etc. on the back part of the grate are pulled over the top of the live coal, after which the latter is spread evenly over the grate, and covered with a light charge of fresh coal.

It is well to allow the fire to burn down somewhat before cleaning, as it will then be easier to clean. The damper should be partially closed, to prevent cold air from rushing into the furnace and cooling the furnace plates above the fire. For this reason, and also to prevent loss of steam pressure, the cleaning should be done quickly. In cleaning a fire, special attention should be devoted to cleaning out the corners near the door; being somewhat inaccessible, they are apt to be neglected, to the detriment of the fire.

36. Banking Fires.—Should it be desirable to have a boiler lying idle, with steam up and the fires ready to generate steam to the full capacity of the boiler at short notice, the fires are *banked*. This is done by first cleaning the fire, and then either pushing the live coal against the bridge or pulling it to the front and covering it with fresh fuel. The fire will lie smoldering, the air supply being regulated, by the damper, the fire-door, and the ash-pit damper, to keep the fire burning sufficiently to keep up the steam pressure. When it is desired to start the fires again, some of the fuel covering the live coal is skimmed off, and the live coal is spread over the grate. The fire-door is then closed, the ash-pit and smokestack dampers are opened, and the fire allowed

to burn up, for 2 or 3 minutes, perhaps, when it is covered with a light charge of fresh fuel.

37. Hauling Fires.—When the demand for steam suddenly ceases, as, for instance, in case of a breakage necessitating stoppage of the engine for a considerable period, the burning fuel has to be drawn from the furnace. The hard labor entailed in *hauling the fire*, as it is termed, may be reduced considerably by allowing the fire to burn low, if the circumstances permit it. When it becomes necessary to haul the fire, in case of the water getting low in the boiler, it is considered the best practice to deaden the fire before hauling by spreading over it a heavy charge of fresh fuel or ashes.

38. Lighting Fires.—To start a fire, the grate is lightly covered with coal. Wood is piled evenly on top of it and covered with some coal. Greasy waste is put in front of the wood. This, when lighted, will soon ignite the wood, the fire gradually working toward the bridge; the layer of coal below the burning wood protects the grate bars from the heat and thus prevents warping of the bars. As soon as the wood is burning freely, more coal is put on. The fire-door should be kept open while the wood is burning, and the ash-pit damper shut. The fire should burn very slowly at first, so as not to injure the boiler by the unequal expansion of its various parts. The rate at which the fire burns may be regulated by the smokestack damper.

39. Practical Hints on Firing.—As it is usually desirable to keep the steam at an even pressure, the following points ought to be observed. Two fires in the same boiler should never be cleaned at once. When putting a fresh charge of fuel into the furnace, do it as rapidly as possible to avoid the inrush of cold air, which will injure the furnace plates and also reduce the temperature of the gases of combustion. Two furnaces in the same boiler should not be charged at once, as the fresh charge deadens the fire and reduces the generation of steam. The usual method of managing the fire is as follows: Let A and B be two single-ended Scotch boilers with two furnaces each, numbered as in

Fig. 2, and suppose that bituminous coal is used. Furnace *1* is charged with fresh fuel; the fire in furnace *2* is leveled off with the hoe; the fire in furnace *3* sliced, and, next, the grate of furnace *4* is cleaned from below; furnace *2* is now charged with fuel, and *3* is leveled, *4* sliced, and the grate of *1* cleaned; furnace *3* is charged, *4* leveled, *1* sliced, and the grate of *2* cleaned; furnace *4* is charged, *1* leveled, *2* sliced, and the grate of furnace *3* cleaned.

It will thus be seen that the same operation is not repeated at the same time in one boiler, and, furthermore, that each operation is performed in the numerical order of the furnaces. Thus, if the first fire sliced were in furnace *3*, the next one to be sliced would be in furnace *4*, the next one in furnace *1*, and the next one in furnace *2*.

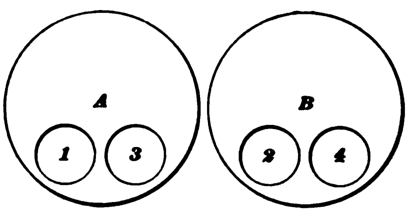


FIG. 2

The furnaces numbered with the even numbers are in one boiler; those with the odd numbers are in the other boiler.

This system of firing is in vogue in sea-going steamers, or wherever the engineer in charge believes in doing work systematically. Of course, the system may be changed to suit varying conditions, the arrangement of the operations as described being intended to furnish an idea as to a systematic way of firing. The same system may be and is followed in cleaning fires. The length of time a fire will burn without needing cleaning depends on the amount of coal burned and the quality of the coal. When consuming about 15 pounds of coal per square foot of grate surface per hour, with average coal, the fire will burn about 12 hours before it must be cleaned.

40. Lumps of coal should not be thrown into the fire, but should be broken into pieces about the size of a man's fist. The fresh charge of fuel should be spread evenly over the fire. With irregular-sized coal, leveling with the hoe has to be resorted to some time before a fresh charge of

fuel is put into the furnace. Should it be noticed, while charging, that the fire has burned out in one spot, owing to clinkers having formed on the grate and prevented admission of air to the fuel above it, or to the burning away of the fuel, the grate at that spot should be cleaned and the open spot filled with live coal before a fresh supply of fuel is put into the furnace. The back of the grate should not be allowed to become bare, as the inrushing cold air will cool the hot gases of combustion and greatly reduce the amount of steam generated. The ash-pit should be kept clean, as ashes accumulating therein will prevent the free access of air to the furnace. The length of time that air is admitted above the fire, and the amount of it, depends on the quantity and quality of the coal. For coal rich in hydrocarbons, a longer time will be required than for other coal. But in no case should air be admitted after the hydrocarbons are expelled from the coal, which, with anthracite, will be in 2 or 3 minutes after charging. By watching the steam gauge, the condition of the fires can be told, and a look at the burning fuel will show whether fresh fuel is needed, or the fires require slicing, or the air supply is insufficient, either by reason of the damper being closed or the grate clogged up.

Before cleaning fires, it is often advisable to increase the feed and to work up the steam pressure, and to reduce the feed while cleaning. A certain quantity of heat is stored up in the extra water fed into the boiler, which is liberated when the feed is reduced, and tends to keep up the steam pressure.

An anthracite fire does not need to be sliced, as no crust is formed on the surface of the fire. Slicing will merely mix the cinders and clinkers with the live coal and deaden the fire. The **T** bar should be run in under the fire to break up the cinders.

COMBUSTION OF OIL

41. Since the first attempt to use petroleum for fuel, early in the history of the oil industry, down to the present time, the principal problem to be solved has been the method of burning. Naturally, the first attempts to burn

oil were by lighting the surface of a mass of oil as it lay in vessels or pans. The amount of combustion that it was possible to obtain with this method was limited because of the limited surface of oil that could be exposed to the flame. Attempts were made to increase this surface by causing the oil to run over plates and broken brickwork, but the great difficulty of supplying air to the entire surface of the oil and the interior of the flame soon caused this method to be given up. Next, attempts were made to gasify the oil and then burn the gas. Part of the heat of combustion was utilized in converting additional oil into gas, and thus it was hoped to render the process continuous. Because of their exposure to the intense heat of the furnace, these various arrangements rapidly deteriorated; also, a carbonized residue of soot and tar was found to clog the pipes and passages. The difficulties of this method were early seen to preclude the possibility of success along this line, and the attention of engineers was turned to the spraying method, with results that leave little to be desired.

42. The spraying method of burning oil consists of introducing the oil into the furnace in a very finely divided state by means of a jet of air or steam and causing the combustion of the oil while in the spray form. The most obvious method in a boiler room of producing a spray of oil is by means of a jet of steam, and consequently it is found that a very large class of oil burners depend on this spraying agent. The method by which spraying is caused to take place varies slightly in different types of burners, but, in general, it may be stated that the oil and steam are caused to mingle within the passages of the burner, and the high velocity of the steam blows the oil into the furnace in a very finely divided state.

A burner that utilizes steam as the spraying agent is shown in cross-section in Fig. 3. It consists of a cylinder *a* having one end closed by means of a cap *b* and the other end by a stuffingbox. Extending axially through this cylinder is a pipe *c* having one end pointed and making a steam-tight

joint in a conically shaped orifice in the cap *b*. The other end of this pipe is closed with a plug *d*. A tapered opening through this plug is closed by means of a needle valve *e*. The pipe *c* can be moved in or out of the cylinder *a* by means of the hand wheel *f*, thus closing or opening the conical orifice in the cap *b*. A hood *g* is cast in one with the cap *b* and serves to protect the end of the burner from the intense heat of the furnace. Oil under pressure enters the burner through the pipe *h*, passes through the needle

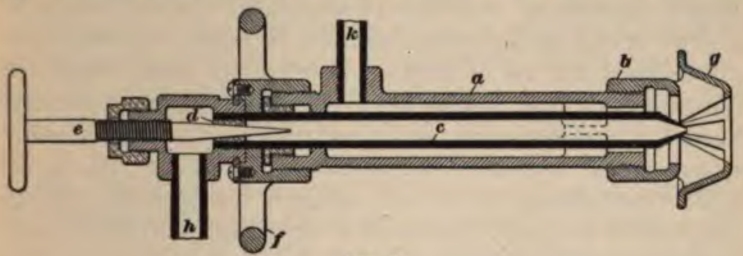


FIG. 3

valve *e* into the pipe *c*, and out through a small opening in the end into the furnace. Steam from the boiler enters through the pipe *k*, surrounds the pipe *c*, and passes through the interior of *a* into and out of the conical opening around the end of *c* into the furnace. The supply of oil is regulated by the valve *e*, and the supply of steam is controlled by turning the hand wheel *f*. The oil mingling with the steam as it passes out of the burner into the furnace is converted into spray, and combustion is readily caused to take place within the furnace.

43. An oil burner that uses air instead of steam as the vaporizing agent does not differ materially from the burner just described. As in the burner using steam, the oil enters a small inner pipe, and the atomizing agent, that is, the air, is supplied through the outer shell of the burner to the tip or cap on the furnace end, where the oil and air mingle and pass into the furnace. The relative efficiency of steam and air has not as yet been definitely determined, but there can be no question that, with air, the mixture of the combustible

elements of the oil and the oxygen of the air is much more intimate at the beginning of combustion than could possibly be the case with steam and oil. Another advantage that air has over steam in oil burners is that there is no loss of steam and consequently of fresh water. This is of great importance aboard ship, where fresh water is very valuable. The United States Navy Department found, as the result of experiments with liquid fuel, that the water consumption of a steam-operated burner is between 4 and 5 per cent. of the total evaporation. Another advantage of the air is that it can be stored under great pressure in tanks, and thus the oil burners can be utilized in getting up steam when all the boilers in the battery are cold. With a steam-operated burner, a coal fire must be started under one boiler in order to get steam to operate the oil-burning devices. The principal disadvantage of the use of air is the necessity of the installation of an air compressor or blower, and the consequent increase in weight.

44. Attempts have been made to produce a burner that is a combination of the two classes and that uses both steam and air for atomizing purposes. Extended tests by the United States Navy show that a burner of this class requires several times as much steam for successful operation as does a burner using steam alone. This class of burner is much more complicated in design, more costly to manufacture, and requires much more skill in operation than burners using either air or steam alone. The Naval Board unqualifiedly condemned burners using both air and steam as the spraying agent.

45. The steam that is used in spraying the oil does not in any manner increase the temperature of the flame, if, indeed, it does not actually decrease it. The water forming the steam is undoubtedly decomposed by the high temperature that obtains in the furnace, thus setting free the hydrogen and oxygen, of which the water is composed. Farther back in the furnace, where the temperature is lower than that necessary for the disassociation of the elements of

water, the hydrogen burns and water is again formed. When water is broken up into its elements by exposure to high temperature, exactly the same amount of heat is absorbed as is given out when the resulting free hydrogen is again burned. The water thus formed passes off as steam with the other products of combustion. One effect of the use of steam under pressure is undoubtedly beneficial, in that its action tends to produce a more even distribution of heat in the furnace, and transfers heat from the point where it is greatest to the back of the furnace and to the tubes, where it is less. A source of loss due to the presence of steam is that it undoubtedly passes up the smokestack at a considerably higher temperature than that with which it enters the burner; the additional heat being, of course, taken directly from the furnace.

46. The accessories that should accompany an oil-burning installation are: oil pumps arranged in duplicate for pumping the oil from the storage tanks and supplying it to the burner under pressure; heating tanks for heating the oil to a high temperature by means of a steam coil; strainers for removing all dirt and grit; and when air is used as the atomizing agent, an air compressor or fan. It is highly important that the oil pumps should be in duplicate, as a breakdown might cause a stoppage of the entire plant. Insurance companies prohibit feeding the oil by the gravity system, owing to the danger of fire. It is also desirable to have the strainers arranged in pairs, as the opening in the burner for the discharge of the oil into the furnace is very small, and very little dirt or grit will materially affect the operation of the burner. It has been found advantageous to arrange for heating the air used in the combustion of the oil. This is usually accomplished by forcing the air over heated surfaces or through pipes that are more or less exposed to the heat of the furnace. Heating the air does not increase the heat of the flame, but tends to promote combustion. It is conducive to the best working of the burner to raise the temperature of the oil to the ignition point as soon as possible after it leaves the tip of the burner,

and this result is found to be much hastened by heating the air supply to as high a temperature as practicable. The pressure of the steam used in the burners may vary between the limits of 20 and 70 pounds per square inch. There have been no tests made public to show the most economical pressure, but the results of the oil-fuel tests of the United States Navy Department show that the higher the pressure the greater the amount of water that was evaporated, and also that the efficiency of the burner slightly increased. Increasing the pressure of the oil and steam had the disadvantage of increasing the steam consumption of the burner. When air is used, the pressure is much lower, varying from less than 1 pound with an ordinary fan blower up to about 20 pounds with an air compressor.

47. It is usually found advisable to make the steam pressure in the burner a certain proportion of the pressure in the boiler, as oil burners are most efficient when the steam used as the atomizing agent is superheated. The cheapest means of superheating, where a superheater is not installed, is by means of free expansion of the steam in a reducing valve; the steam pressure in the burner should be about 25 per cent. of the pressure in the boiler. As any change in the pressure of the air, steam, or oil makes a readjustment of all valves necessary, care should be taken that the pressures in an oil burner be kept as nearly constant as possible when once the burner is in operation. Too much stress cannot be laid on this point. It is advisable to have the steam supply of the burners so arranged that any boiler can furnish steam to any burner.

48. In the following description of the method of raising steam on a battery of boilers having an oil-burning installation using steam as an atomizing agent, it is assumed that there is steam in one boiler for operating the burners. If there is not, steam must be raised by means of a coal fire under one boiler for that purpose.

Steam is first turned on to a burner, and any accumulation of water in the steam pipes is thus blown out. Next, the oil valve is slowly opened, and the oil spray that is thus

formed is ignited by means of oily waste or, preferably, by a short-handled torch. The supply of steam and oil should now be regulated until the furnace is completely filled with a flame that is steady, white or bluish white, and that gives off little or no smoke at the top of the smokestack. Much smoke is a certain indication of a poorly adjusted burner. As the experience of the fireman increases, he will be enabled to judge of the efficiency of combustion by the sound that is emitted from the furnace. If, for any reason, the flame is extinguished at any burner, the oil supply should be immediately shut off, or the accumulation of oil vapor in the furnace will cause a more or less violent explosion when the burner is relighted. When steam is used as the atomizing agent, and a burner is extinguished, it is advisable to shut off at once both steam and oil; but when air is used, and a burner is extinguished, it is well to shut off the oil only. When the air is also shut off, the compressor or fan causes a larger quantity to pass through the other burners, and, as this disturbs the proper relation between the quantities of oil and air, much smoke is produced and a readjustment of the valves on all the burners is thus made necessary. All this is obviated by permitting the air to blow through the idle burner until it is expedient to relight it. Care should be exercised that the oil is not heated in the oil heater to so high a temperature that volatile gases are driven off and caused to accumulate in any of the pipes. This gas, if it is forced through the burner, interrupts the steady flow of oil, and thus will often extinguish the burner. If air becomes mixed with this gas, as it may in some forms of oil heaters, the gas and air will burn within the pipes and necessitate a shut down. Burners have been known to get red hot from combustion within the oil passages. The strainers, arranged in duplicate, should be regularly cleaned out at stated intervals, instead of waiting until one is completely choked and the supply of oil is thereby diminished. There should not be any oil allowed to appear on the floor of the fireroom. Firing with oil does not require bodily strength, nor experience with firing coal.

DRAFT

49. Natural Draft.—The difference between the weight of a column of hot gases contained within a smokestack and the weight of an equal column of cold air results in an upward motion of the hot gases within the stack, which motion is called **natural draft.** It is well known that any gas, when heated, is lighter, bulk for bulk, than when cool. Now, when the hot gases pass into the smokestack they have a temperature of 400° F. or 500° F., while the air outside the smokestack has a temperature of from 40° F. to 90° F. Roughly speaking, the air weighs twice as much, bulk for bulk, as the hot gases. Naturally, then, the pressure in the smokestack is a little less than the pressure of the outside air. Consequently, the air will flow from the place of higher pressure to the place of lower pressure; that is, into the smokestack through the furnace. As an example, suppose that a smokestack is 150 feet high and that the temperature of the hot gases is 500°. A column of gas at this temperature, 150 feet high, and of 1 square foot cross-section, weighs about $6\frac{1}{2}$ pounds. A column of air at 60°, of the same length and cross-section, weighs about $11\frac{1}{2}$ pounds. Hence, the difference in pressure at the bottom of the chimney is $11\frac{1}{2} - 6\frac{1}{2} = 5$ pounds per square foot. This difference in pressure is spoken of as the **draft pressure.**

50. It is customary to express the pressure of the draft in inches of water. It has been shown that the pressure of the atmosphere, 14.7 pounds per square inch, supports a column of water 34 feet high. 34 feet of water = 14.7 pounds per square inch; or, $34 \times 12 = 408$ inches of water = 14.7 pounds per square inch; = 2,116.8 pounds per square foot. Therefore, 1 inch of water $= \frac{14.7}{408} = .036$ pound per square inch; $= \frac{2,116.8}{408} = 5.2$ pounds per square foot.

The draft pressure, or intensity of the draft, is measured by means of a water gauge, one form of which is shown in

Fig. 4. As inspection shows, it is a glass tube open at both ends, bent to the shape of the letter **U**; the left leg communicates with the smokestack; the air outside the smokestack, being heavier, presses on the surface of the water in the right leg and forces some of it up the left leg; the difference in the two water levels H and Z in the legs represents the intensity of the draft and is expressed in inches of water.

The draft pressure required depends on the kind of fuel used. Wood requires but little draft, say $\frac{1}{2}$ inch or less; bituminous coal generally requires less draft than anthracite. To burn anthracite, slack, or culm, the draft pressure should be $1\frac{1}{4}$ inches of water.

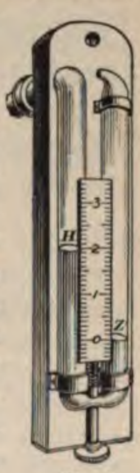


FIG. 4

51. The area of the smokestack must be such that the gases of combustion may be discharged freely. Experience has shown that an area of 1 square foot of smokestack to every 8 square feet of grate surface is a fair ratio, representing the practice of some of the best builders. The intensity of the draft depends on the height of the smokestack; hence, the amount of coal that may be burned per square foot of grate surface per hour may be roughly estimated from the height of the smokestack, provided that the area of the smokestack is of the required size. The height of the smokestack is to be taken as the perpendicular distance between its top and the grate. Owing to coals of the same kind varying in chemical composition, and between wide limits, the quantity of coal burned with a given smokestack and grate area cannot be estimated with any great exactness; this must be borne in mind when applying the rules given.

The probable maximum rates of combustion attainable under natural draft are given by the following rules, which have been deduced from the experiments of Isherwood, United States Navy,

where W = weight, in pounds, of coal burned per square foot of grate area per hour;

H = height of smokestack, in feet.

Rule I.—*To find the amount of anthracite that may be burned per square foot of grate surface per hour under the most favorable conditions, subtract 1 from twice the square root of the height of the smokestack.*

Or, $$W = 2\sqrt{H} - 1$$

Rule II.—*For anthracite burning under ordinary conditions, subtract 1 from one and one-half times the square root of the height of the smokestack.*

Or, $$W = 1.5\sqrt{H} - 1$$

Rule III.—*For best semianthracite and bituminous coal, multiply the square root of the height of the smokestack by 2.25.*

Or, $$W = 2.25\sqrt{H}$$

Rule IV.—*For ordinary soft coals, multiply the square root of the height of the smokestack by 3.*

Or, $$W = 3\sqrt{H}$$

The maximum rate of combustion is thus fixed by the height of the smokestack; the minimum rate may be anything less.

EXAMPLE.—What is the maximum coal consumption per hour of a vessel fitted with six boilers with two furnaces each, the length of the grate being 6 feet, the width 3 feet 6 inches, and the height of the smokestack 65 feet? Ordinary soft coal is being used.

SOLUTION.—Using rule IV,

$$W = 3\sqrt{65} = 24.187 \text{ lb. per sq. ft. of grate area per hr.}$$

The total grate area $= 6 \times 2 \times 6 \times 3.5 = 252$ sq. ft. Hence, the maximum coal consumption $= 252 \times 24.187 = 6{,}095.12$ lb. per hr. Ans

52. Mechanical Draft.—As previously explained, natural draft is caused by the upward flow of heated air. The force of the draft depends a great deal on various conditions, such as the direction and force of the wind, the temperature of the air, the height of the smokestack, the thickness of the fire, etc. To make the draft independent of any of these conditions, various mechanical arrangements may be used and a draft thus created mechanically. In that case

the draft is spoken of as **mechanical draft.** In marine work, mechanical draft is applied in one of three ways: Either the air is forced by a fan into an air-tight fireroom, when the system is called the *closed fireroom system*, or the air is forced by suitable mechanism into air-tight ash-pits, when the system is called the *closed ash-pit system*. Both of these systems are spoken of as *forced-draft systems*. In the third application of mechanical draft, a partial vacuum is created in the uptake or base of the smokestack by fans or steam jets; this system is spoken of as an *induced-draft system*.

In naval vessels, the closed fireroom system of forced draft is extensively used. All openings from the fireroom are closed air-tight, and air forced in until the desired pressure has been reached. This system has not found much favor in the mercantile service, and the closed ash-pit system has been adopted instead.

53. Howden's closed ash-pit system, in which hot air under pressure is delivered into the ash-pits and furnaces, is shown in Fig. 5. By means of a blower located in a suitable place, cold air is forced through the pipe *H* into a closed air chamber located in the uptake immediately above the doors of the front connection. The gases of combustion pass through numerous vertical tubes, shown at *A*, contained in the air chamber, thus heating the air surrounding the tubes. The heated air passes into an air-tight reservoir *B*, attached to the front end of the boiler and surrounding the furnaces, as well as the front connection *G*. This reservoir, projecting about 10 inches from the front of the boiler, receives the air under pressure. The air admitted above the fire enters through the valve *C* into a space between the outer and inner furnace doors, both of them swinging on one hinge. The inner door is perforated, and is provided with a perforated air-distributing box *D*, the outer door serving to retain the air pressure. The air passing through the valve *C*, besides filling the spaces between the doors, also fills the space around the furnace door, whence it passes into a perforated air-distributing box *E*, covering the whole surface of

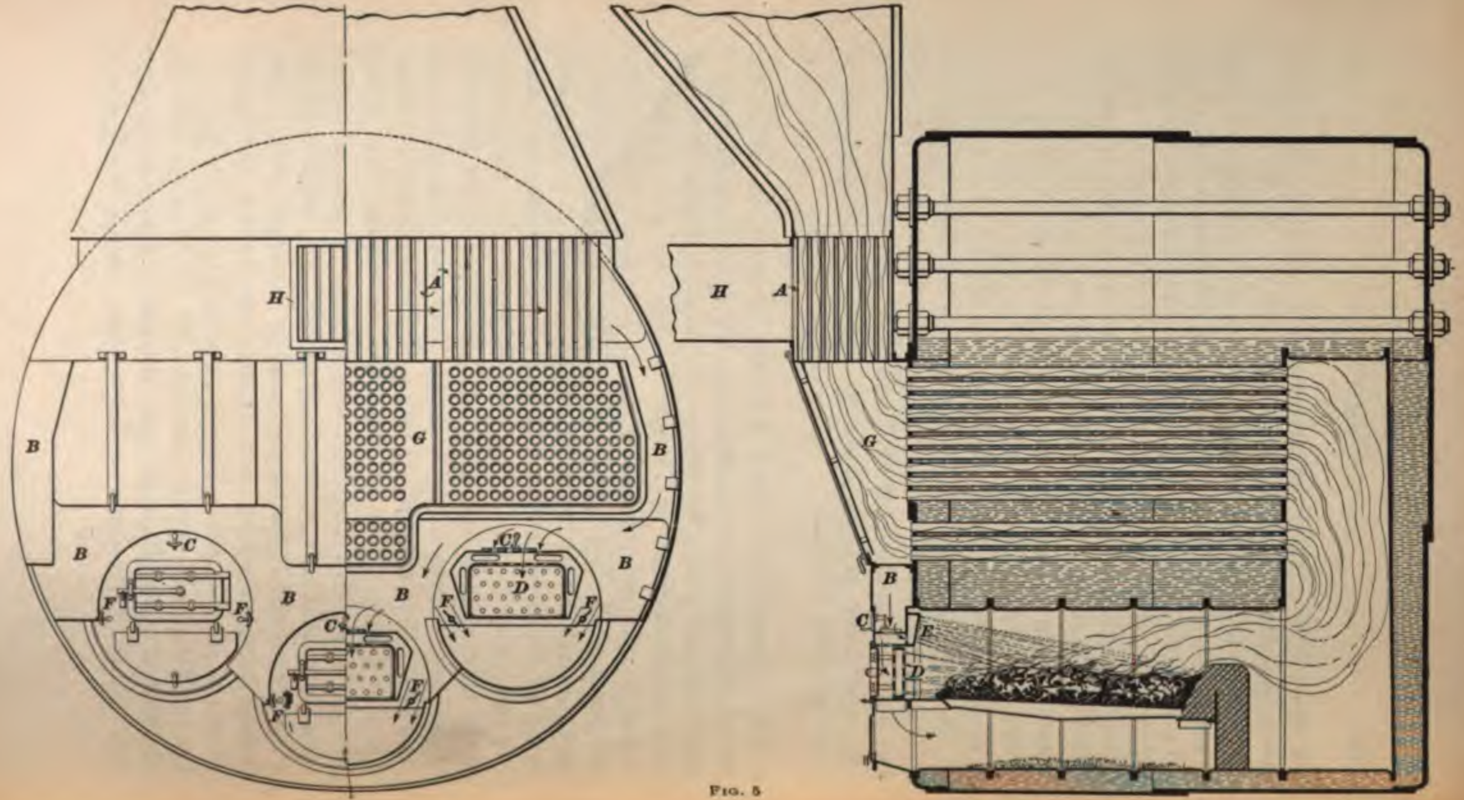


FIG. 5

the furnace front inside the furnace. Part of the air passes through the perforated dead plate into the ash-pit, which is closed by hinged doors. The rate of admission of air to the ash-pit is regulated by means of the valves F.

In operation, the rate of combustion of the fuel is governed by the valves F regulating the air pressure. The valves C are adjusted at the beginning of the trip to suit the character of the fuel used. The jets of highly heated air admitted by these valves above the fire tend to promote complete combustion. In some closed ash-pit systems, cold air is delivered under pressure into the ash-pit.

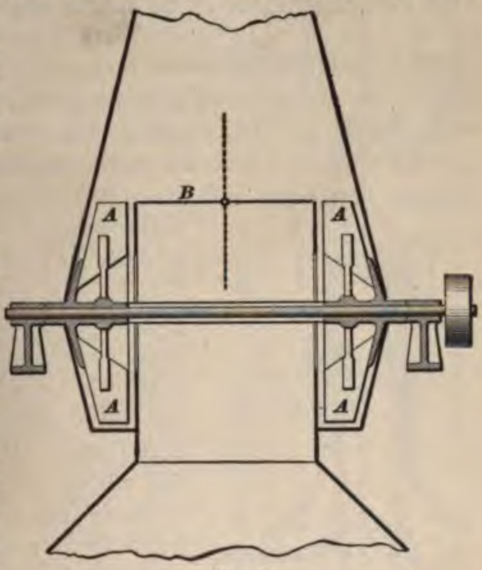


FIG. 6

54. The application of a fan to the base of a smokestack, in order to induce draft, is shown in Fig. 6. A pair of fans A, A are located at the base of the smokestack, one on each side. Both fans are mounted on the same shaft, which passes through the smokestack. The shaft is belted to a steam engine, not shown in the figure. In operation, the damper B is closed while using the exhaust draft. The rapidly revolving fans exhaust the air in the uptake, thus reducing the pressure in the tubes and furnaces, and thereby producing a more rapid current of air through the furnaces, both below and above the grate. No alteration of the arrangement of the furnace fittings is necessary. This system can be applied very cheaply to steamers already built.

55. In the **Ellis & Eaves induced-draft system,** a fan is placed outside the uptake; the fan inlet is below and the fan outlet is above the smokestack damper. This arrangement allows either natural or mechanical draft to be used at will. The air entering the furnace is heated by passing around nests of tubes through which the gases of combustion pass before reaching the fan; the furnaces are closed off from the fireroom by air-tight furnace doors and ash-pit doors, and suitable mechanism is provided for admitting air either above or below the grate, as may be required.

56. Induced draft by means of a steam jet is to be found in many American steamships, and has much to recommend it. The chief points in favor of the steam jet are the ease with which it can be applied to existing boilers, its low first cost, and the reliability of its action; besides, being located inside the smokestack, it does not occupy valuable space. While its steam consumption is greater than that of the fan engine used in mechanical-draft installations, it affords a ready means of increasing the evaporative power of a boiler or boiler plant. Whether it will increase the efficiency of the boiler, that is, increase the number of pounds of water evaporated per pound of coal, is somewhat doubtful.

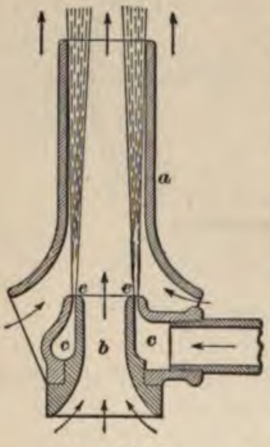


FIG. 7

In installing a steam jet, it must not be overlooked that the steam used is lost, while with a fan engine the steam can be, and usually is, exhausted into the condenser, thus saving the fresh water. This consideration limits the steam jet chiefly to vessels running in fresh water, and to such other vessels in which the saving of the fresh water is not essential.

The construction of a **Bloomsburg steam jet** is shown in Fig. 7. It consists of a casing *a* having a central nozzle *b*

at its lower end. Steam from the boiler enters the chamber c formed by the lower part of the casing and the nozzle, and flows through the annular opening at e. The steam issues from this opening at a high velocity and thus induces a current of air to flow in the same direction as the steam. A number of these small jets are attached to a spider, which consists of a central casting with radiating steam pipes, the ends of each steam pipe carrying a jet. Its arrangement in the smokestack is shown in Fig. 8. A steam gauge is attached to the system of jets, showing the steam pressure in the jets. To start the induced draft, simply turn on the steam; to regulate the intensity of the draft, open or close the valve.

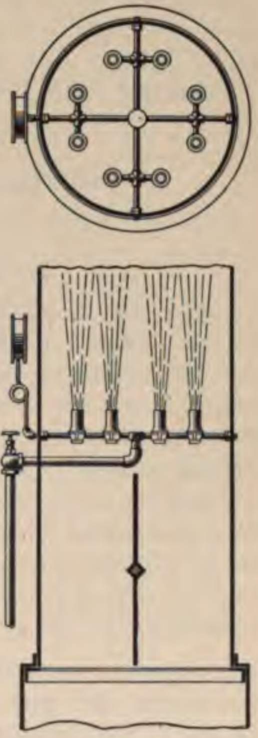


Fig. 8

57. With non-condensing engines, the exhaust from the engine may be turned into the smokestack in order to induce draft; this arrangement is quite common on steamers navigating the western rivers of the United States of America and on those engaged in similar service elsewhere. In that case, it is usual to provide a valve by means of which the exhaust can be discharged either directly into the atmosphere or into the smokestack.

58. Air may be forced into a closed ash-pit by means of a steam jet, which is similar to the device described in Art. **56.** While this application of forced draft is quite common in stationary practice, it is very uncommon in marine work. However, it has an advantage in the case of coal that must be used continuously and that is liable to clinker badly, adhering firmly to the grate bars and sides of the furnace. In this particular case, it seems to be the consensus of opinion

among engineers that the admission of steam into the ash-pit, incidental to the employment of the steam jet, has a tendency to alleviate the clinkering and subsequent sticking of the clinkers to the bars.

ECONOMIC COMBUSTION

ECONOMIC COMBUSTION OF COAL

PRINCIPLES OF COMBUSTION

CONSTITUENTS OF COAL

1. Introduction.—Since fuel, chiefly in the form of coal, is the source of all the energy made available by the steam engine, a study of the principles governing the economical generation of power by means of the steam engine must begin with a study of the combustion of fuel, and of the means of preventing waste of heat by guarding against those conditions that tend toward incomplete combustion. Heat lost by imperfect combustion cannot be recovered; no arrangement of elaborate and economical machinery in the engine room will utilize the energy wasted in the fireroom through ignorance or carelessness on the part of the fireman.

2. Definitions and Classification.—Bituminous coal is composed largely of various compounds of carbon and hydrogen called **hydrocarbons.** When the coal is heated, these compounds are driven off, partly in the form of permanent gases and partly as vapors that may easily be condensed or changed to a liquid form. The process of separating the gases and vapors from the part of the coal that cannot be vaporized by the mere action of heat is called **distillation;** the substances driven off from the coal by heat are called **volatile substances;** while the portion remaining forms

coke, which is composed chiefly of carbon. This carbon is called the **fixed carbon** of the coal.

The volatile substances may be divided into two classes, viz., *non-combustible* and *combustible* substances. The first class consists mostly of water, free oxygen, and nitrogen; these are driven off when the coal is heated—the water as steam and the gases in their free state. The second class consists of the hydrocarbons, which comprise numerous compounds of hydrogen and carbon. When the coal is heated, a part of the hydrocarbons is driven off in a gaseous form and a part as vapors.

The principal gases in the volatile combustible are **carbureted hydrogen,** or **marsh gas** (consisting of 1 atom of carbon and 4 atoms of hydrogen, as shown by its symbol, CH_4), and **olefiant gas,** C_2H_4. With many coals, free hydrogen is given off in considerable quantities; small quantities of other less important gases also are generally present.

The vapors are mostly coal tar and naphtha, with small quantities of sulphur. Their presence can be detected by inserting a cold iron bar into the yellow gases rising from a fresh charge; a sticky coating, consisting mostly of the condensed tar, will form on the cold metal.

The proportion of the volatile matter in coal depends on its composition. Anthracite consists almost entirely of fixed carbon and ash; in some bituminous coals, the greater part is volatile. The relative proportions of fixed gases and condensable vapors in the volatile parts of coal also vary with the composition of coal. In some cases, the volatile matter contains considerable quantities of the tarry vapors and heavy hydrocarbon gases, as C_2H_4, while in others it consists largely of the light marsh gas, CH_4, and free hydrogen.

The quantity and composition of the volatile matter depend not only on the composition of the coal itself, but also on the conditions under which distillation takes place, a difference in the temperature and the presence or absence of air or steam modifying the composition of the vapor and gases to a very great extent. Irregular firing and draft

result in a great difference in the quantity and composition of the gas burned in the furnace at different periods.

With few exceptions, coal contains small quantities of sulphur, usually in combination with some other element; one of the most common compounds is that of sulphur and iron, known as **iron pyrites.** When the coal is heated, the sulphur is separated from the iron and burns to sulphur dioxide, SO_2. The heat derived from the combustion of the sulphur found in coal is small, but the sulphur dioxide formed, in combination with the moisture in the gases, corrodes iron very rapidly; any relatively cold metal exposed to the gases from coal rich in sulphur is rapidly corroded and destroyed.

COMBUSTION OF CONSTITUENTS OF COAL

3. Combustion of Volatile Constituents.—At the temperature generally existing in a boiler furnace, the tar and other liquids vaporize and mix with the gases; this gaseous mixture is readily burned under proper conditions of air supply and temperature. The conditions involved in the combustion of the gases and vapors may readily be studied by the aid of the flame of a common tallow candle. The tallow forms a supply of solid hydrocarbon that, under the action of the heat of the flame, is liquefied; then, by capillary attraction, it is drawn up into the wick to a point where the heat is sufficient to vaporize it. A process of distillation here goes on, and a blue, transparent, cone-shaped mass of vapor and gas, shown at *c*, in Fig. 1, is formed. The heat of the flame causes a current of air to approach from all sides and provides a supply of oxygen that comes in contact with the surface of the hot cone of vapor. The air that reaches this part of the flame, however, is not sufficient for the complete combustion of the vapor, and since the hydrogen in

FIG. 1

the hot cone has a greater affinity for oxygen than it has for carbon, the hydrogen combines with all the available oxygen and burns to water, H_2O. The carbon is left in the form of minute solid particles that become highly heated and give this part of the flame its bright yellow color. As the hot particles of carbon rise, they come in contact with the inward current of air, which furnishes oxygen sufficient to burn them to CO_2. The gas so produced, mixed with the H_2O from the luminous portion *b* of the flame, forms the nearly colorless outer and upper section *a* that gradually mixes with the surrounding air, cools, and becomes wholly invisible.

If the end of a glass tube is inserted into the inner cone of the flame, as shown in Fig. 2, a part of the gas that has not yet received a supply of air may be drawn off. In passing through the tube, this gas is cooled below the temperature at which it will burn and issues from the tube into the air without igniting. By applying a lighted match to this gas and heating it, it may be lighted and burned. This simple experiment illustrates two of the most important principles involved in the economical combustion of the gases in a boiler furnace. It shows: (1) that the gas that has been cooled below the temperature of ignition cannot be burned merely by furnishing a supply of air; (2) that when the gas is supplied with air, it can be burned if its temperature is raised to the igniting point.

FIG. 2

4. Ignition Temperature.—In order to burn the fixed carbon that is in the coal, a high temperature is needed to cause the atoms of carbon to combine with the oxygen supplied by the air. The igniting temperature of the fixed

carbon, and of the volatile substances also, is estimated to be about 1,800° F. Since the maximum temperature in the furnace rarely exceeds 2,500° F., and is ordinarily several hundred degrees less, it is seen that, on account of the relatively small difference between the igniting temperatures of the carbon and gases and the maximum temperature in the furnace, constant care is needed to prevent the temperature in the furnace falling below 1,800° F. The temperature in the furnace can be judged quite accurately by the appearance or color of the fire, in accordance with the relation between color and temperature given in the accompanying table.

When the supply of air is sufficient and the temperature high enough, the carbon burns to carbon dioxide, CO_2, which, being the product of complete combustion, is incombustible. With a high temperature and a deficient air supply, carbon monoxide, CO, is formed. Since this gas is the product of incomplete combustion, it can be burned to CO_2 by bringing it into intimate contact with air while highly heated.

TABLE I

COLOR AND TEMPERATURE OF FIRE

Temperature Degrees Fahrenheit	Appearance	Temperature Degrees Fahrenheit	Appearance
980	Red—just visible	2,010	Dull orange
1,290	Dull red	2,190	Bright orange
1,470	Dull cherry red	2,370	White heat
1,657	Full cherry red	2,550	White, welding heat
1,830	Bright red	2,740	White, dazzling heat

5. Combustion of Solid Carbon.—When solid carbon burns on a grate, the chemical changes or reactions are about as follows: Some of the oxygen of the air that rises through the grate combines with the first layers of hot carbon in the proportion of two atoms of oxygen to one atom of carbon, and carbon dioxide, CO_2, is formed. As the gases rise

through the fire, more of the oxygen combines with carbon, and as long as the supply of air is sufficient and well distributed, the combination is mostly in the proportion that produces CO_2. With a thick bed of fuel, however, or an arrangement of the fuel that does not permit of a proper distribution of the air, there will be some portions of the fire in which the supply of oxygen is not great enough to furnish the two atoms for the production of CO_2; only one atom of oxygen will be available for combination with some of the carbon atoms burned, and the product will be CO. Further, when a molecule of CO_2 comes into close contact with the hot carbon, the attraction of the carbon for oxygen is so great that one atom of the oxygen leaves the CO_2 and combines with an atom of carbon; two molecules of CO are thus formed, one by the separation of one of the oxygen atoms from the molecules of CO_2, and the other by the combination with an atom of carbon of the oxygen atom so released. The separation of a given weight of CO_2 into CO and O absorbs as much heat as was developed when the CO combined with oxygen to form the CO_2; the net production of heat is the same, whether a certain amount of carbon is burned to CO directly and passes off in that form, or a part of it is first burned to CO_2 and this gas is then decomposed, with the production of CO and of oxygen that combines with the remainder of the carbon to form CO. The carbon monoxide formed in the fuel bed passes into the furnace, and if there is not sufficient oxygen present, or if the temperature is not high enough, it will pass away unburned. If sufficient oxygen is present and the furnace temperature is high enough, each molecule of CO will combine with another atom of oxygen and thus burn to CO_2.

6. Summary.—A careful study of the outlines of the processes involved shows that economical combustion, both of the volatile matter and of the solid carbon, involves the following essential conditions: (1) there must be a supply of air sufficient to furnish the oxygen required for complete combustion; (2) this air must be so distributed as

to bring the oxygen into intimate contact with all parts of the fuel; (3) the temperature must be high enough to bring about the combustion. With any of these essentials lacking, there will be incomplete combustion and a loss of heat.

SMOKE

SMOKE FORMATION

7. The smoke arising from an ordinary coal fire may properly be divided into two classes: (1) the partly condensed tarry vapors produced by the distillation of the fuel; (2) the minute particles of solid carbon left when the hydrocarbons are but partially burned. Smoke of the second class is so objectionable that some steamships, in order to promote the comfort of passengers, burn only anthracite, which burns practically without smoke.

8. When hydrocarbons are heated in the presence of air, the affinity of the hydrogen for oxygen is great enough to cause it to separate from the carbon and combine with the oxygen. If the supply of air is sufficient and the temperature is sufficiently high, the carbon set free will burn and practically no smoke will be produced. If, however, there is a limited supply of air or too low a temperature for their combination with oxygen, the carbon atoms will combine with each other and form molecules of carbon that collect into minute particles of solid carbon. It is the presence of these heated carbon particles that gives color to the flame of an ordinary lamp or fire. If they can be supplied with oxygen at a high enough temperature, they will burn, as in the flame of the candle, Fig. 1. Under unfavorable conditions, however, these solid particles of carbon do not burn, but are deposited as soot or are carried out of the furnace with the gases as smoke. The nature of these unfavorable conditions can well be studied by considering the action of an ordinary kerosene lamp. With a good chimney, a flame not too high, and a clean burner, the lamp burns with a

bright, steady flame and no smoke. The bright flame indicates a high temperature, and the absence of smoke shows that the air supply is ample and well distributed. Insert a cold iron rod into the flame from the top of the chimney; a deposit of soot—unburned carbon—collects on the rod, and the gas that rises along the sides of the rod is cooled so much that a part of the carbon is not burned and considerable smoke is formed. Interfere with the air supply by partly closing the top of the chimney or the holes in the burner; a cloud of smoke is formed. Remove the chimney, so as to interfere with the distribution of the air to the lower part of the flame while a large volume of cold air meets the upper part, and a condition obtains in which the carbon is cooled by the action of an excessive supply of cold air imperfectly distributed. Turn the wick too high, and the formation of the gases is too rapid in comparison with the supply of air; hence, part of the gases cools below the temperature required for perfect combustion.

SMOKE PREVENTION

9. It is seldom that the supply of air to a boiler furnace is not great enough to burn the carbon and prevent the formation of smoke; in fact, it is often found that large volumes of smoke are accompanied by a liberal supply of oxygen. The more common condition when smoke is formed is too low a temperature of the fire or a distribution of the air that prevents oxygen from reaching the carbon before it is cooled by contact with the boiler plates. Heavy firing, by means of which large volumes of gas are formed and the furnace greatly cooled, is one of the most prolific causes of smoke production. Owing to the high ignition temperature of solid carbon, smoke, when once formed, is extremely difficult to burn. With properly constructed furnaces, good draft, and careful management, **smoke prevention** is possible; *smoke consumption*, however, may be said to be practically impossible under any of the conditions existing in the furnace of a steam boiler.

10. If the bed of burning fuel is thick and the draft sluggish, carbon monoxide will rise from the surface of the fire, and if it is not burned to carbon dioxide by mingling with air admitted above the fire, it will pass through the flues and cause a great loss of heat—about 10,100 British thermal units for each pound of carbon.

Two remedies for this type of loss are available: (1) the fuel bed may be made so thin that sufficient air can enter through the grate, either to prevent the formation of carbon monoxide or to burn it to carbon dioxide as soon as it is formed; (2) sufficient air to burn the carbon monoxide that rises from the coal may be admitted into the furnace above the fire. If there is a brisk fire and a consequent high temperature in the furnace, the carbon monoxide can be easily burned by either of the two methods given, a short, pale blue flame being produced near the surface of the fuel. With a slow, dull fire, the temperature in the furnace may be too low to ignite the escaping gas, which will pass out with any air that may be present, no combination taking place. There is, thus, a double loss; heat is wasted not only by the escape of carbon monoxide, but also by the heating of the excess of air.

11. The presence of unconsumed carbon monoxide is not easily detected by mere observation, owing to the gas being colorless. By regulating the fire, however, so that a high temperature is maintained in the furnace, and by watching the surface of the burning fuel to see that sufficient air is present to produce the pale blue flame previously mentioned, it is possible to make reasonably sure that the amount of unconsumed carbon monoxide will be exceedingly small.

With a large grate area and a very low rate of combustion, there is usually formed a considerable quantity of unconsumed CO, owing to the relatively low temperature in the furnace. In the case of a single boiler, the remedy is to restrict the grate area by bricking up with one or more courses of firebrick, so as to secure a higher rate of combustion and the consequent high temperature. In the case of

several boilers, one or more may be cut out of service when conditions permit in order to secure a higher combustion rate and temperature in the furnaces of the boilers in service. Simply admitting more air when there is a very low rate of combustion is liable to aggravate the heat losses, since it will result in still further lowering the temperature of the furnace.

The escape of unconsumed hydrocarbon, with its attendant loss of heat, is more readily detected by direct observation of the fire than carbon monoxide, especially when the loss from this source is large. The yellowish vapors that rise from freshly fired coal are familiar to all firemen and are an evidence of unconsumed hydrocarbons. When the heavier hydrocarbons are contained in the smoke in considerable quantities, the smoke from the smokestack is yellowish in color and has a tarry odor.

12. The conditions required for the complete combustion of the hydrocarbons are the same as for the fixed carbon of the fuel; that is, a sufficient air supply, an intimate mixture of the air with the gases, and a high temperature. In practice, it seldom happens that a large part of the carbon in the volatile matter escapes unburned, even when a great deal of black smoke is produced. Owing to its finely divided state, a small quantity of carbon will give a high color to a large volume of gas coming from a chimney, and there will appear to be a serious waste when the actual heat loss is really comparatively small.

Numerous careful tests in stationary steam-engineering practice have shown that the production of black smoke does not necessarily represent a great loss in efficiency. In many cases, it has even been found that a reduction of efficiency accompanied the use of furnace arrangements that were successful in preventing smoke. The reason for this appears to be that the gases were burned with a large excess of air, which, while so controlled as to burn the carbon to CO or CO_2, thus rendering it invisible, carried much more heat from the furnace than was developed by the comparatively small

weight of carbon that would have appeared in the smoke. The losses due to solid carbon falling into the ash-pit, the escape of unburned hydrocarbons, incomplete combustion of the fixed carbon on the grate, and a needless excess of air are always much more serious than the loss due to the formation of black smoke.

Although smoke in itself represents no serious loss of heat, its production in large quantities is often the result of conditions that produce imperfect combustion of the carbon monoxide and the hydrocarbons. A low furnace temperature, the piling of coal on the fire in such large quantities that an excessive volume of combustible gas is given off, while, at the same time, the furnace temperature is lowered by the heat absorbed in the process of distilling the volatile matter from the freshly fired coal, and also the admission of large volumes of cold air, tend to produce black smoke, and, besides, represent serious and unnecessary heat losses.

AIR SUPPLY TO FURNACE

AIR SUPPLY BELOW GRATE

13. It has been shown that the complete and economical combustion of coal and the prevention of smoke depend primarily on a sufficient air supply being brought, under proper conditions, into intimate contact with the fixed carbon and volatile combustibles. Without the proper distribution of the air supply, the high temperature necessary for complete combustion cannot be maintained. One of the best methods of supplying the air is that in which the fire is thin and open enough to permit sufficient air to rise through the grate and fuel bed. The advantages of this method are as follows:

The air, in rising through the fuel bed, becomes highly heated. If a clean, even fire is maintained, the air supply is well distributed and comes in intimate contact with the gases almost as soon as they are formed; the air and gases are thus thoroughly mixed in the vicinity of the hottest part of

the furnace, and complete combustion follows. The danger of chilling the boiler plates by a current of cold air is less than when the air is admitted at any point above the grates.

With this method of air supply, the firing must be carefully done; the fire must be maintained at a moderate and even thickness, the grates must be kept clean, and each air space must be kept as free from clinkers as possible. No bare spots should be allowed to form, and, finally, the coal should be supplied in small quantities and often, each shovelful being spread over as much surface as possible. Large lumps should always be broken before being fired, and in no case should a thick mass of freshly fired coal be allowed to collect in any part of the furnace. Such a method of firing demands care and close attention on the part of the fireman, but if it is carefully followed, the coal will be burned in a very economical manner and without the formation of black smoke. The reduction in the amount of coal that must be handled will, in a great degree, make up for the apparent increase in labor connected with such a system of firing.

The thickness of the bed of fuel to be used with this system of firing depends on the quality of the fuel and the intensity of the draft. In general, it may be stated that the best results will be obtained with a good strong draft that will permit of a fuel bed, with reasonably good bituminous coal, of from 7 to 12 inches in thickness. The thickness can best be determined by actual trials and carefully watching the results.

AIR SUPPLY ABOVE GRATE

14. In a great many cases, it is impracticable, or at least difficult, to regulate the fire so carefully as to secure a proper supply of air through the grates. For example, in Western river and similar service, where the demand for steam and the intensity of the draft are very irregular, it is found to be practically impossible to maintain a depth of fire that will conform to the irregularity in the conditions of running. The heavy currents of air drawn through the thin bed of fuel by the irregular action of the exhaust tear holes in the fire if

it is kept too thin, thus allowing large quantities of cold air to enter at one place; these currents of cold air, in addition to their evil effects on the combustion of the gases and their chilling effect on the boiler, often carry considerable quantities of solid fuel into the flues.

It is seldom that the conditions are as unfavorable in seagoing service as in Western-river service, but there are many cases in which the irregularity of the draft, or of the demand for steam, is considerable; the quality of the fuel may also make it very difficult to keep a clean, open fire that will permit the steady supply of air demanded; it is, therefore, often essential that means be provided for furnishing, through openings above the grate, at least a part of the air required to burn the gases. Of the methods used for this purpose, the one most commonly found is, perhaps, either a partly opened fire-door or some special arrangement of openings through the fire-door that will serve as an inlet for the air and distribute it over the fire in a more or less satisfactory manner.

A partly opened door must be regarded as one of the most unsatisfactory means of getting air into the furnace that could be devised. The air enters in a large stream that is more likely to cool the small proportion of the gas that it reaches below its ignition temperature than it is to promote its combustion; further, the concentrated current of cold air is almost sure to strike a limited section of the furnace, which section becomes chilled; this, in turn, produces severe stresses in the boiler plates. Much more satisfactory results are obtained when the door is provided with special openings that serve to divide the entering current of air and distribute it over a considerable part of the grate. The perforated inner door is particularly useful in this respect; it serves the combined purpose of protecting the outer door from the heat radiated from the fire and of dividing the entering current of air into a large number of small jets, which are considerably heated in their passage over its surface, and are then distributed to the fire through the perforations and around the edges of the plate.

15. With a large furnace working under natural draft, it may be difficult either to admit sufficient air through the perforated door to supply all parts of the furnace or to properly distribute the air so admitted. To remedy this defect, recourse is had, in marine practice, to admitting the air above the grate under pressure, as is done in the Howden and also in the Ellis & Eaves mechanical-draft systems. In firebox and locomotive-type marine boilers working under natural draft, a few rows of hollow staybolts may be used just above the surface of the fire; these admit air in small jets that enter that part of the furnace least likely to be reached by air from the door, at right angles to the current of the gases, and the air is thus more easily mixed with the gas than would be the case if the currents were parallel. A point that would seem to be unfavorable to the ultimate success of any attempt to introduce air through the sides of the furnace in this manner is the difficulty of controlling the quantity so introduced in accordance with the varying conditions of draft and fuel supply.

16. The admission of air through properly constructed openings in the bridge wall and above the fire has been shown, by experiment, to produce economical results when a bituminous coal rich in volatile matter and producing a long, smoky flame is used. The openings through the bridge wall should be so arranged as to discharge the air in a number of jets at a right angle to the direction of flow of the gases. The space back of the bridge should then be large enough to form a combustion chamber in which there will be a thorough mixture and complete combustion before the air and gas are chilled by entering the tubes. A large space produces a relatively moderate velocity of flow of the gases, which is favorable to the intimate mixture of the air and gas.

HEATED-AIR SUPPLY

17. While it would be an advantage to have the air enter at a high temperature, no means have yet been devised that will accomplish this in an economical manner with

natural draft. Of course, the temperature of the air is raised to a slight degree in its passage through the ash-pit, or the openings in the setting leading to the bridge, and in passing through the bridge to the openings through which it is discharged, but the actual gain by this means is relatively unimportant, even when rather elaborate systems of passages are used. Some of the difficulties attendant on this method are the following: In order to prevent loss from too much or too little air, the passages must be controlled by dampers that should be regulated according to the condition of the fire and the rate of combustion; this demands close attention and intelligent care on the part of the fireman, who must either maintain nearly constant conditions in the furnace or change the dampers as often as there is a material change in the condition of the fire. The passages through the bridge wall may give trouble by becoming clogged with ashes and clinkers that have been pushed back from the fire during cleaning or carried back by the draft, and so become useless. The method is limited in its application to furnaces so constructed that there is room for a combustion chamber of considerable size, in which mixture and combustion may take place before the gases are cooled.

The difficulties just mentioned are minimized in mechanical-draft systems, like the Howden and the Ellis & Eaves systems, where the heated air is discharged, under pressure, where it will do the most good, and where there is an absence of passages likely to give trouble by becoming choked with ashes and refuse. The proper control of the supply of heated air above and below the grate demands intelligent supervision, however, in both of these systems.

FURNACE AND COMBUSTION CHAMBER

SHAPE AND SIZE OF FURNACE

18. The lowest temperature at which ignition of the gases can take place is about 1,800 F. By reference to the Steam Tables, in *Heat and Steam*, it is seen that the temperature of the water in the boiler, even under a pressure of 200 pounds per square inch, is less chan 400° F.; this is practically the temperature of the surface of the boiler. It is therefore evident that any gas that comes into close contact with the boiler before being burned will be cooled below the point of ignition, and, unless subsequently heated, will be carried to the smokestack unconsumed. With coals containing large quantities of volatile matter and burning with a long smoky flame, it follows that the firebox must be of ample depth to provide a space in which the great volume of gas can burn before being cooled; or that there must be a combustion chamber in which the gas can burn after leaving the firebox.

The danger of loss from cooling the gases too suddenly is greatest in internally fired boilers of the vertical, locomotive, and firebox types. Unless the crown sheet is unusually high above the grate, vertical boilers are especially unfitted for the use of rich bituminous coals. Locomotive-type and firebox boilers, if properly managed, are better; by the use of the coking system of firing, the gas must pass through the length of the firebox and over the hot bed of coke, thus giving it considerable time to burn before entering the tubes, the heat radiated from the coke helping to keep it at a temperature at which combustion is possible.

Firebox and locomotive-type marine boilers may be fitted with firebrick arches that extend from the tube sheet toward the door and force the gases to take a path, first toward the door and then back above the arch to the tubes. The arch becomes highly heated, thus preventing the cooling of the gases before they become mixed with the air, and the path

of the gases is lengthened sufficiently to enable them to burn before entering the tubes. It also increases the life of the flues by preventing the entrance of large volumes of cold air when the door is opened, and thus aids in maintaining a more even temperature.

The opinion of railroad men, almost without exception, is in favor of the firebrick arch for locomotives burning bituminous coals, and there is no doubt that the principle involved in its use can be employed to great advantage in marine work for similar boilers.

ADMISSION OF STEAM INTO FURNACE

19. When steam is admitted into the furnace through the ash-pit, or otherwise, it is decomposed by the heat into its constituents, hydrogen and oxygen. This process, however, absorbs as much heat as can possibly be developed by the combustion of the hydrogen thus formed. From this it follows that there can be no possible gain in heat from introducing steam. In fact, there are several features that may cause the use of steam to result in an actual loss of heat; there is danger that much of the hydrogen liberated by the decomposition of the steam may escape unburned owing to the reduction of temperature of the fuel bed produced by the decomposition of the steam. Also, the steam enters the ash-pit at a temperature seldom above 212° F. and passes into the chimney at the temperature of the flue gases, which is rarely below 400° F. It thus carries more heat into the chimney than it introduces into the furnace.

The use of a steam jet for forcing air into the ash-pit, and the admission of steam through a small pipe to the ash-pit are quite common in stationary practice, but very uncommon in marine work. There are cases, however, where either method may be advantageously employed in marine work, which occur when coals that tend to clinker badly and stick to the grate bars must be used. With such coals, the effect of the steam is to prevent the clinkering to a considerable extent; this enables the fireman to keep the grate cleaner,

prevents the destruction of the grate, and, on account of the improved condition of the fire, permits of a better distribution of air and more complete and economical combustion. In the case of a steam jet forcing air into the ash-pit, or a steam jet in the smokestack, using live steam, there is often an increased economy in the use of the coal over that which is obtained by natural draft; the gain, however, must be ascribed solely to the improved air supply, and not to the fact that steam is used. A similar improvement in the draft by means of a higher smokestack or by some mechanical-draft appliance will generally give even better results than can be obtained by the use of steam.

COMBUSTION CHAMBER

20. In internally fired boilers of the Scotch marine and similar types, having large furnace flues that open into a chamber in the rear, the gases that are cooled by contact with the walls of the flue and pass through it unconsumed sometimes burn in the rear chamber, which, for this reason, is given the name **combustion chamber.** With externally fired boilers of the flue type, the space back of the bridge serves the purpose of a combustion chamber to a certain extent. The gases, however, tend to rise and flow along the cold surface of the boiler; it is, therefore, difficult to prevent a considerable body from thus becoming cooled below the ignition temperature and passing unconsumed into the tubes.

Water-tube marine boilers generally have no distinct combustion chamber, unless the term be applied to the upper space within the jacket surrounding the boiler. The gases rise nearly vertically from the fuel bed and pass from the firebox immediately into contact with the tubes; the narrow spaces between the tubes divide the gases into thin sheets that are rapidly cooled below the temperature of ignition.

The vertical direction of the current of gases in the furnace makes it difficult to secure any considerable admixture of air from the fire-door; the chief dependence for the air supply

must therefore be on the air that rises through the grates and passes upwards through the bed of fuel. These conditions make it essential, for complete and economical combustion, that a sufficient supply of air be admitted through the grate itself, and that the supply be well distributed over the whole grate area, so that it may become mixed with the gases almost as soon as they are formed. It is also important that the grate be placed far enough below the tubes to permit of a thorough mixture of the gas and air and of complete combustion of the gas before it enters the space between the tubes.

HEAT LOSSES AND THEIR PREVENTION

MISCELLANEOUS HEAT LOSSES

21. A portion of the heat generated in the furnace is usefully expended in evaporating water, but a large percentage of the heat is often wasted. Some of these heat losses are unavoidable, while others are due to poor management or poor design of the boiler.

The heat losses due to incomplete combustion of the carbon and hydrocarbons, the formation of smoke, excessive air supply, etc. have already been pointed out, and the methods of preventing them have been explained. In addition to these losses, there is the loss of heat inseparable from the use of natural draft. Since the draft depends entirely on the difference in density between the gases within the smokestack and the air surrounding the smokestack, it follows that the heat required to produce the difference in density (that is, to produce the draft), while not available for the generation of steam, is essential to the operation of a boiler; while the heat thus expended may be called a heat loss, it is nevertheless an unavoidable loss.

Some heat is lost by radiation from the boiler itself and some from the connections. This loss, while it cannot be prevented entirely, can be minimized by covering the exposed parts with some good non-conducting material.

MOISTURE AND DISTILLATION HEAT LOSSES

22. There are processes accompanying the combustion of coal in the boiler furnace that in themselves absorb great quantities of heat and, in consequence, have an important bearing on the question of complete and economical combustion. All moisture that enters the furnace with the coal must be evaporated at the expense of the heat developed by the combustion of the coal; the vapor thus formed passes out of the smokestack at a temperature seldom less than 400° F. Assuming that the moisture enters the furnace at a temperature of 70° F. and leaves at 400° F., 1,200 British thermal units, nearly, will be lost for each pound of moisture in the coal. Moisture in coal can only be driven off by heating it above ordinary temperatures, but it will be readily absorbed at ordinary temperatures; hence, it is important that coal when stored is not exposed to rain—it should always be stored under cover. In some cases, it may be advantageous to wet bituminous coal; when wet, especially if the coal is fine, it cokes better, and hence there is less waste from coal falling into the ash-pit. Wetting is recommended only for bituminous slack and anthracite culm, and should be as moderate as will secure the results desired. It would be much better, however, when the draft is moderate, to use grates having smaller openings. When the draft is exceptionally strong and very fine coal is burned, wetting becomes almost a necessity. If not done, the strong draft will actually carry a large percentage of the fine coal up the smokestack.

The distillation of the volatile matter is a process that, with all bituminous coals, and to a lesser degree with anthracite, absorbs a great deal of heat, as is shown by the drop in the steam pressure when a large quantity of fresh coal is thrown on the fire; this drop is largely due to the lowering of the furnace temperature through the heat absorbed by the distillation of the volatile matter. While the absorption of heat is necessary to drive off the volatile combustible, the losses attendant upon a lowering of the furnace temperature can be minimized by frequent light charges of coal.

HIGH-TEMPERATURE HEAT LOSSES IN SMOKESTACKS

23. In many cases, there is a large amount of heat passing out of the smokestack in excess of that required to produce the necessary intensity of draft. In practice, it has been found that, when the temperature of the escaping gases has been lowered to about 500° F., the draft will be ample. If their temperature is in excess of this, it usually indicates a serious heat loss. The high temperature may be due to several causes, either singly or combined, among which may be mentioned insufficient heating surface, inefficient heating surface, and poor water circulation.

It is rather hard to decide where to place the blame in case the temperature of the escaping gases is excessive. In general, the trouble is that the efficiency of the heating surfaces has become impaired by reason of the collection of soot and condensable tarry vapors on the fire side, and the deposit of scale on the water side. The obvious remedy is to clean the surfaces, and to clean them thereafter at such intervals as will keep them in a state of high efficiency.

If the heating surfaces are clean and the temperature of the escaping gases is excessive, the trouble, with fire-tube boilers, may be due to poor circulation. It is difficult to state just exactly what should be done to improve the natural circulation, since boilers vary so much in design. In general, it is cheapest to use some suitable apparatus designed to give a forced circulation. In flue boilers and water-tube boilers, the circulation is usually free and strong; an excessive temperature of the escaping gases with these types of boilers is usually due to dirty or insufficient heating surfaces.

Insufficiency of heating surface is generally found in cases where, due to the exigencies of service, boilers are forced beyond the steam-making capacity for which they were installed. This calls for an increased combustion rate per square foot of grate surface, in consequence whereof an increased volume of gas passes through the tubes and over the heating surface at a higher velocity. Since the transfer of heat from the heated gases to the water depends to a

large extent on the time during which they are in contact with the heating surfaces, a proportionately smaller amount of heat per pound of gases is transferred to the water. A partial remedy for loss due to this cause is to lengthen the time the gases are kept in contact with the heating surfaces, as may be done by using spiral retarders in the tubes of fire-tube boilers; or, if feasible, by fitting suitable baffle plates between the tubes of water-tube boilers, in order to lengthen the path of the gases. Another partial remedy is a feedwater heater placed in the path of the waste gases before they pass out of the smokestack.

RULES FOR BOILER EFFICIENCY

24. The **efficiency** of a boiler is the ratio of the difference between the heat in the steam delivered by the boiler and the heat in the feedwater to the heat that would be developed by the perfect combustion of the fuel, and is expressed by dividing the former quantity by the latter. Thus, if a test shows a total supply of heat of 270,187,000 British thermal units, and a useful application of 186,429,030 British thermal units to the evaporation of water into dry steam, the efficiency of the boiler is $\frac{186,429,030}{270,187,000} = .69$.

Boiler efficiency thus determined consists of two factors not readily separable—the efficiency of the furnace as a heat producer and the efficiency of the boiler as a heat absorber. It is possible to have a furnace so well constructed and managed that the combustion is nearly perfect, and still have a low efficiency of the boiler as a whole, owing to inefficient heating surfaces, large radiation losses, etc. In order to secure a high efficiency of the boiler, as a whole, it is necessary to pay strict attention to each and every detail and to keep it in the most perfect condition possible.

25. Having seen that complete, i. e., economical, combustion depends on a sufficient air supply intimately mixed with the combustibles, and a high furnace- and combustion-

chamber temperature to insure ignition, the following rules will be self-evident:

1. Fire light and often.
2. Keep the fire as thin as circumstances will permit.
3. Keep the fire clean.
4. Keep the space between the grate bars clear.
5. Keep the ash-pit clear.
6. When using bituminous coal, use the coking firing system, if possible.
7. Regulate the draft so that it will be strongest when a fresh charge of coal is put into the furnace.
8. Do not let the fire burn out in spots.
9. Do not let the fire burn too low before charging.
10. If possible, fire at regular intervals and in regular charges.

MARINE-BOILER FEEDING

(PART 1)

FEED-APPARATUS

ARRANGEMENT

INTRODUCTION

1. The **feed-apparatus** of marine boilers comprises suitable machinery for forcing the water into the boiler, piping for conveying it to the boiler and delivering it therein at the desired place, and suitable valves for regulating the flow of the feedwater. Auxiliary devices often used in connection with the feed-apparatus are feedwater heaters, feedwater purifiers, and circulation-improving devices intended to better the transfer of heat from the burning fuel to the water in the boiler; and in cases where the steam used in the engine is condensed and returned to the boilers without being mingled with the water used for condensing it, there must be some means of making up for the loss of feedwater caused by leakage.

The machinery used for forcing the water into the boiler consists either of *pumps* or of *injectors*, the former being either driven directly by the propelling machinery or independent steam pumps. Usually, two independent sets of feed-apparatus are fitted to marine boilers, both of which may have pumps or injectors, or one set may comprise pumps and one set injectors.

In practice, the feedwater for marine boilers is obtained either by using the surrounding water, which method is

today used ordinarily only by vessels navigating fresh water, or by condensing the steam used in the engine and, without mingling it with the water used for condensing, returning it to the boiler, which method is used by practically all ocean-going vessels.

2. There are two methods of condensing the steam used by the engine. In the first method, the steam is condensed by being brought in direct contact with a spray of cold water in a vessel called a **jet condenser.** The condensing water usually flows to the condenser by gravity and atmospheric pressure. The mingled condensing water and condensed steam are drawn from the condenser, together with any air present, by the **air pump;** as much of this water as is required for the boilers is supplied to the feed-pumps, the remainder being discharged overboard by the air pump. It is obvious that by this method the impurities contained in the condensing water are carried into the boiler. In the second method, the steam is condensed by being brought into contact with metallic surfaces kept cold by water pumped over them, which are contained within a vessel called a **surface condenser.** The condensed steam, vapor, and any air present is removed from the condenser by the air pump and delivered into a vessel called a **hotwell,** from which the condensed steam is taken by the feed-pump and returned to the boilers. The condensing water is pumped through the condenser by the **circulating pump.** In a surface condenser, the condensed steam and the condensing water do not mingle, and hence the water taken from the hotwell, unless contaminated by leakage or the deliberate admixture of sea-water, is pure distilled water, generally free from harmful ingredients, except oil or grease. The grease or oil in the feed-water is usually extracted by filtering, although in some cases attempts are made to remove it from the exhaust steam before condensation.

3. It is of the utmost importance that the feed-apparatus of a marine boiler shall be as perfect as possible, not only in regard to its capacity to supply the boiler with all the

water it may require, even in case of a heavy leak, but also that the possibility of its becoming inoperative shall be reduced to a minimum. The pipes should be run as straight as convenience will permit, avoiding all unnecessary bends and turns, and they should be so located that they may be readily traced from end to end. Although it is sometimes necessary to place some of the pipes under the engine-room and fireroom floors, it is better to avoid this whenever possible; but when it must be done, the floor plates over them should be laid so that they can easily be taken up and thus access be gained to the pipes underneath, particularly at those points where valves are located.

INSTALLATION

4. In sea-going steam vessels, every pump in the ship is usually provided with the pipe connections necessary to allow it to be used for feeding the boilers. Fig. 1 illustrates one way in which the feedpipes may be connected; for the sake of clearness, only the pipes relating to the feed-apparatus are shown. The boiler is shown at *C*, the engine at *D*, the condenser at *E'*, the air pump at *F*, and the circulating pump at *G*. The plunger feed-pumps *A*, *A'* are worked by the engine, taking their water supply through the suction pipes *o*, *o'* from the hotwell *B*, and discharging the water through the main feedpipe *b* into the feed check-valve *C*, whence it passes through the stop-valve *D'* into the boiler. The feed check-valve is a device that permits the feedwater to flow into the boiler, but automatically prevents water in the boiler flowing into the feedpipe.

The piping just described constitutes the **main feed.** On all marine boilers, a bronze or brass-seated stop-valve must be attached between the boiler and each feed check-valve to facilitate access to the check-valves while the boiler is under steam, and also as an additional safeguard to keep the water in the boiler. Check-valves will sometimes fail to seat properly, allowing the water in the boiler to leak past them while the pump is stopped, the water passing through the

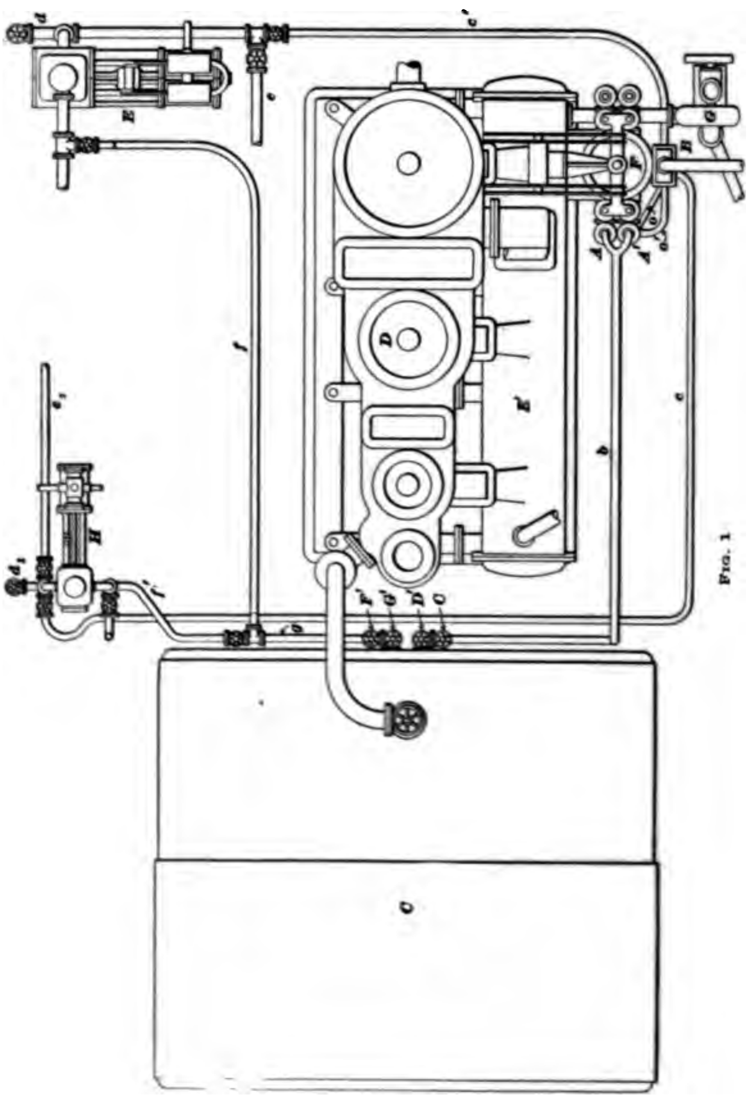


Fig. 1

the main feedpipe into one of the other boilers in which the pressure may be somewhat less, thus allowing the water to become dangerously low in the first boiler. By closing the stop-valve in the feedpipe this is prevented. The duplex steam pump E and the single-acting pump H are connected to the hotwell B by the suction pipes c' and c. The pump E is provided with branch suction pipes d and e leading, respectively, to the sea and to the fresh-water ballast tank. Similar suction pipes d_1 and e_1 are fitted to the pump H, and other branch suction pipes leading to other available sources of water supply may be fitted to each pump, as required. The pump E delivers through the pipe f, and the pump H through the pipe f', into the donkey feedpipe, or auxiliary feedpipe g, whence the water passes through the donkey check-valve F' and the stop-valve G' into the boiler. The two feed-arrangements just described constitute the **donkey feed,** also called the **auxiliary feed.**

It has been shown that in the arrangement described there are three ways of feeding the boiler, each independent of the others. The main feed-arrangement may be used, or, if desired, either of the other two arrangements. Hence, the possibility of being unable to supply the boiler with feed-water is very remote.

5. The point at which the feeder enters the boiler varies with different builders. In shell boilers, some take the feed through the front head of the boiler, some through the rear head, and some through the shell. The common practice is to lead the feedwater to the coolest part of the boiler. This is done by a pipe secured to the inside of the boiler shell. When a large body of comparatively cool water is discharged on to a hot plate, severe local strains are set up in the plate, due to the contracting of the part cooled by the entering feedwater. These strains soon destroy the plate; to obviate this, the end of the internal feedpipe is often closed and the entering feedwater distributed over a large area by numerous holes drilled into the pipe in such a position that the water will issue in a direction away from the

plates composing the boiler. In Scotch boilers, in the best modern practice, the feedwater is discharged by a pipe lying on a level with the top row of tubes, the pipe being perforated and discharging downwards between two nests of tubes.

CONSTRUCTION

PUMPS

6. Kinds of Pumps Used.—The **pumps** employed regularly for pumping feedwater into the boilers are usually of the plunger pattern, and may be either single-acting or double-acting. They are either driven directly by the main engine, in which case it is the usual practice to employ single-acting pumps, or they are independent, in which case they are frequently duplex and double-acting—by duplex being meant that there are two separate pump mechanisms combined to form one pump. Independent single or duplex direct-acting steam feed-pumps are made either vertical or horizontal, the choice of pattern usually depending on the room available. In steamboats navigating the western rivers of the United States of America, a special form of independent steam pump is largely used, in which the pumps are operated by a beam rocked back and forth by a steam engine.

7. Single-Acting Plunger Feed-Pump.—One form of construction of the type of plunger feed-pump generally used in connection with vertical marine engines and driven from one of the crossheads through a rocking beam giving the plunger a to-and-fro motion, is shown in Fig. 2, in which *A* is the plunger, made of Muntz metal. It is cored out to lighten it, and it passes through a stuffingbox and gland lined with white metal. The pump chamber *B* is bolted to the side of the air pump. A valve chamber *C* containing the suction valve *D* and delivery valve *E* is bolted to the pump chamber. These valves work in gun-metal bushings fitted to the valve chamber. Above each valve is a yoke *d*, *e* that prevents the valve from rising too high. A small petcock *K*

itted between the suction and delivery valves; this may used to test the working of the pump and also to admit to the air chamber. The air in the air chamber will ner or later be absorbed by the water, in which case the

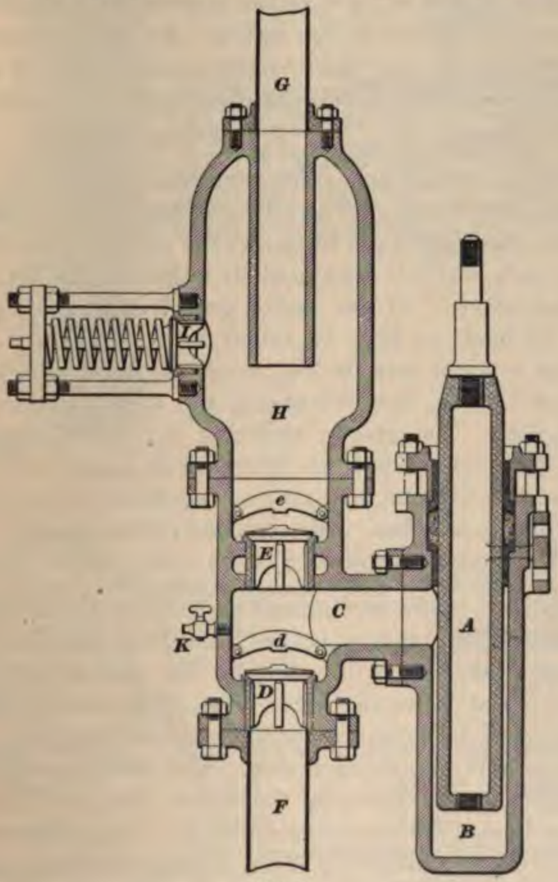


FIG. 2

ves will seat with a considerable shock. Opening the pet-k will admit air on the up stroke of the plunger. On the vn stroke, water will issue if the pump is working prop-y. In some cases, no petcock is fitted. When the valves t heavily, or *slam*, as it is called, loosening the gland of

the plunger will admit some air and cure the slamming. Bolted to the bottom of the valve chamber is the suction pipe *F*, which connects with the hotwell, and is provided with a stop-valve, not shown in the figure. For the purpose of examination, the valve chamber is provided with a removable cover, not shown in the figure. An air chamber *H* is fitted to the top of the valve chamber, and attached to this air chamber is the feedpipe *G* leading to the boiler.

8. The plunger pump *A*, Fig. 2, is working as long as the engine is running, and, consequently, is feeding the boiler or boilers continually. When two or more boilers are supplied from the same main feedpipe, the amount of feedwater entering each boiler is regulated by adjusting the lift of the feed check-valves. If one boiler gets less than its proper amount of feed, as may be found by watching the water gauge, the amount may be increased by giving a higher lift to the check-valve, thus increasing the area of the passage to the boiler. Conversely, reducing the lift of the valve reduces the feed. When an independent pump is used for feeding one boiler, the feed may be regulated by varying the speed of the pump; but, with attached pumps having a constant speed, the check-valve must be used for that purpose.

9. Doctor.—The feed-pumps and heaters of a western-river steamer are shown in Fig. 3. This combination is known as a **doctor**, is in common use, and is located on deck and hence above the water level. The doctor consists essentially of a beam engine, with crank and flywheel, operating four pumps, two on each side. The *lifting* pumps *A*, *A'* are single-acting; they draw the water from the river and discharge it into the open heaters *B*, *B'*. Here the water is heated by being brought in direct contact with the exhaust steam, the exhaust from each engine passing through its own heater. The *force* pumps on the other side take the water from the heaters and force it into the boilers; one of these pumps is shown at *C'*. The base plate *D* on which the doctor is erected contains the various passageways forming the water connections between the pumps, the passages being

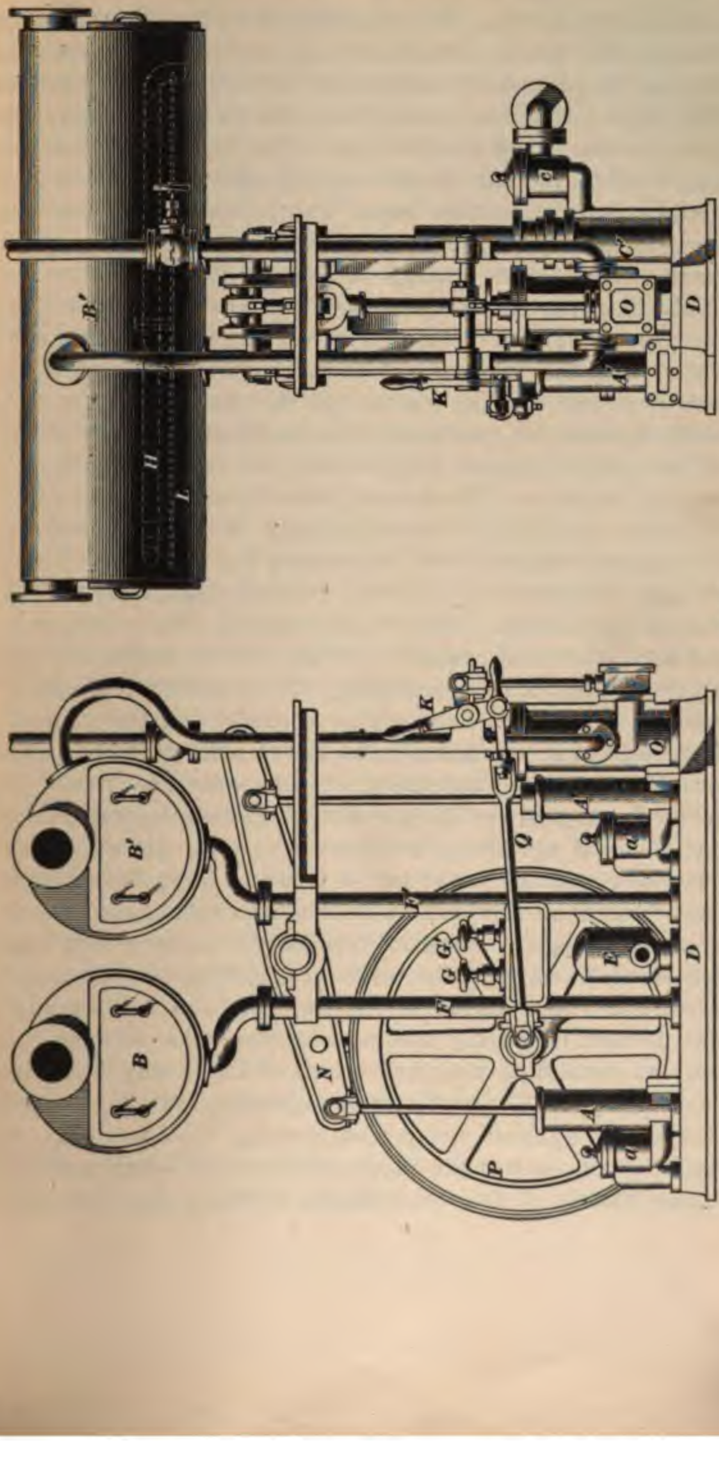


Fig. 3

cored in the casting. The suction pipe to the river connects directly with the *vacuum chamber* E, so called since a partial vacuum is necessarily maintained therein. The object of this vacuum chamber is to prevent shocks and to steady the flow of water in the suction pipe. The lifting pumps being single-acting, the column of water ascending the suction pipe would, if such chamber were not provided, be suddenly brought to rest by the closing of the suction valve of the pump. The kinetic energy of the moving body of water would be given up suddenly, and be expended in a violent blow or shock; but with a vacuum chamber the column of water, with the exception of that comparatively small portion of it between the piston and the chamber, is not suddenly checked, but continues to move during the down stroke of the pump plunger, compressing the rarefied air in the vacuum chamber. Shocks are thus obviated and the inflow of water steadied. A cored passage in the base connects the vacuum chamber with the suction end of the two lifting pumps. The water is delivered through the delivery casings a, a', which contain the delivery valves, into passages in the base plate that connect with the hollow columns F and F', the column F communicating with a and the column F' with a'. The water does not pass straight up the columns, however, but is led through the valves G, G', and then back into the columns, and thence to the heaters. When the doctor is stopped for the purpose of opening and examining the valves of the lifting pumps, valves G, G' are closed and serve to retain the water in the heaters. The heaters consist of wrought-iron shells with cast-iron heads; the exhaust from each engine enters its own heater through one head and leaves it through the other head. The exhaust comes in contact with a coil H of copper pipe near the bottom of each heater; the lifting pumps force the water through this coil and discharge it at the bottom of the heater below the diaphragm L. While this diaphragm does not in any way prevent the exhaust steam from coming in contact with the water, it acts as a baffle plate, preventing violent agitation of the surface of the water in the heater, and at the same

time provides a quiet spot for the collection of floating impurities. An overflow pipe is attached to the heater at or about the level of this diaphragm, which not only prevents the flooding of the heater, but also serves to carry off the oil and other light impurities floating on the surface of the water. The heated water flows by gravity down hollow columns opposite *F* and *F'* to the suction side of the force pumps, which are single-acting plunger pumps. It now passes through the delivery chambers, one of which is shown at *c*, into two pipes that unite to form the main feedpipe. The exhaust pipes from each heater are usually united to form one main exhaust pipe, extending from near the heaters to near the back end of the boilers. Here this pipe forms two branches, each branch leading to one of the smokestacks. Provision is usually made for turning the exhaust either into the smokestack for the purpose of increasing the draft or directly into the atmosphere by dividing the pipe into two branches and placing a rotary valve at the junction. The main feedpipe passes through a stuffingbox into the after end of the **Y** fitting at the after junction of the two exhaust pipes; it then passes through the whole length of the main exhaust pipe and emerges through a stuffingbox at the forward end, where the exhaust is again divided. The main feedpipe then leads downwards and branches off to each boiler. By passing the feedwater through the heaters and exhaust pipe, it is heated to a high temperature.

The pumps are all attached to the walking beam *N*, connected to a steam cylinder *O* at one end and to a crank and flywheel *P* at the other end. The slide valve of the engine is operated by a small crank, or occasionally an eccentric, on the end of a flywheel shaft. To allow the cylinder to be readily warmed up when starting, and also to facilitate the starting of the doctor, the eccentric rod *Q* is hooked over a pin of the bell-crank *K*. When the rod is unhooked from this pin, the slide valve can be operated by hand.

10. While the doctor may appear complex at first sight, it is really a very simple machine, in which the working parts

are simple in construction and very accessible. In western-river service, where the water is frequently very muddy and must always be pumped against very high pressures, the doctor has proved economical and efficient, and has held its own against direct-acting steam pumps and injectors. The general demand for lightness and simplicity in machinery is well met by the manner in which the frame is made use of in providing the water passages, and in the design illustrated by making the steam pipe and exhaust pipe of the steam cylinder serve as crosshead guides.

FEED-VALVES

11. The check-valve fitted to the main feedpipe is called the **main check-valve,** and the one fitted to the auxiliary feedpipe is called the **auxiliary,** or **donkey, check-valve.** The donkey check-valve does not differ in any respect from the main check-valve; it is merely so named from its location.

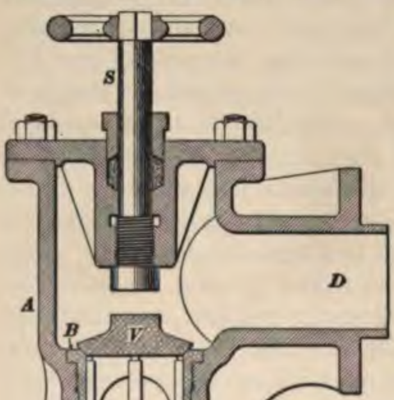


Fig. 4

A common construction of a feed check-valve is shown in Fig. 4. It consists of a body or casing *A* bored out at the lower end to receive the gun-metal bushing *B* forming the seat, and a guide for the valve *V*, which is provided with four wings, as shown. The feedpipe is bolted to the flange shown at *C*, while the passage *D* connects with the boiler. When water flows through the feedpipe into the passage *C*, it raises the valve and flows through the annular opening between the valve and seat into passage *D*, and thence through a stop-valve (not shown in the figure) into the

boiler. The valve *V* allows the water to pass only in one direction; should the direction of the flow of the water be reversed, the valve will return to its seat and shut off communication with the passage *C*. By means of the screw *S*, the height of the lift of the valve can be regulated, thus increasing or diminishing the quantity of water passing through the valve. The screw *S* is commonly made of Muntz metal, to prevent corrosion.

12. Should either the check-valve *C* or stop-valve *D'*, Fig. 1, be closed for some reason while the pump *A* is working, either the pump or the feedpipe will burst. To prevent this, a relief valve that will open and allow the water to escape in case the pressure on the pipe exceeds the pressure on the boiler by, say, 15 pounds per square inch is fitted either to the pump chamber or to the air chamber. The usual construction of a relief valve is shown at *I*, Fig. 2. The area of this valve is equal to that of the feedpipe. Relief valves are sometimes called **safety feed-valves.** The valve is kept to its seat by the spring shown, which abuts against a yoke fitted over two studs provided with nuts, by means of which the tension of the spring can be adjusted to the desired pressure.

INJECTORS

13. Action.—The **injector** is an apparatus for forcing the feedwater into a steam boiler. It was invented in 1858 by an eminent French scientist, Henri Giffard, and was introduced into the United States in 1860 by William Sellers & Co., of Philadelphia.

On investigating the action of the injector, it will be found that dry steam at a given pressure enters the apparatus, passes through several contracted passages, raises several check-valves, and then forces water into the boiler against a pressure equal to that which the steam had when it first began the operation. The steam, in forcing the water through the injector and into the boiler, gives up its heat and performs actual mechanical work as truly as though the

steam acted on a piston and moved a pump plunger with it. Before the action of an actual injector can be studied properly, it is necessary to have a clear understanding of the fundamental principle on which its action is based. This may be stated thus: A current of any kind, be it steam, air, water, or other matter, by reason of friction has a tendency to induce a movement in the same direction of any body with which it may come in contact. The steam entering an injector and moving with an extremely high velocity first carries the air inside the injector with it and thus creates a partial vacuum, causing the water to flow into it. The steam then imparts a portion of its velocity to the water and gives it sufficient momentum to throw open the check-valves and enter the boiler. By striking the cold mass of water, the heat and velocity of the steam will be greatly reduced, and it will be condensed at the same time.

FIG. 5

14. The essential parts of an injector are shown in Fig. 5, which does not represent a practical injector, but serves to illustrate the combination of the essential parts by means of which the injector performs its function. Steam is admitted from the boiler through the steam pipe *a*; the chamber *b* connects to the water supply through the nozzle *c*. The tube *d* is called the *combining tube;* the space *e* is the *overflow space;* the overflow is carried away by a pipe attached to the overflow nozzle *g*. The water passes through the check-valve *i* into the discharge pipe *h* and thence into the boiler. The check-valve may not be part of the injector itself, but it is an essential part of the installation.

The action of the instrument is as follows: Steam is admitted through *a* and flows through the nozzle with a high velocity, passes through the combining tube *d* and out through the overflow *e* and nozzle *g*. This current of steam carries the air in the chamber *b* with it, thus forming a partial vacuum; the pressure of the atmosphere then forces water from the supply into the chamber and into the combining tube *d*. In *d* the steam and water are combined, with the result that the steam imparts a great deal of its velocity to the water and at the same time is condensed. This forms a jet of water that flows from the combining tube *d* with such a high velocity that it passes over the overflow *e* and into the discharge pipe *h*, the energy in the water being great enough to overcome the pressure in the boiler. The water thus flows past the check-valve *i* into the boiler. When the injector is working properly, all the steam that is used to give the water its high velocity is condensed, thus leaving a steady, unbroken jet of water that flows across the space between the combining tube and the discharge pipe. If the water is too hot to condense the steam, or if there is so much steam that it is not all condensed in the combining tube, the steam, owing to its lightness, will not be carried into the boiler, but will flow out into the overflow space *e* and be discharged from the overflow nozzle *g*. This escaping steam breaks the jet of water and interferes with the action of the instrument so much as to stop the flow into the boiler, and serves to show the engineer when there is too much steam admitted for the water that is used. When the supply of steam is too small, the velocity of the jet of water flowing from the nozzle is so small that its momentum is not sufficient to carry it into the discharge pipe against the pressure in the boiler, and the water is therefore discharged through *e* and out of the overflow nozzle *g*. This shows the engineer that the supply of steam is too small.

15. The temperature at which water will be delivered by an injector to the boiler depends on the steam pressure and on the quantity of water being delivered per pound of steam,

the feedwater temperature remaining constant. Thus, if an injector is worked at its maximum capacity with a steam pressure of 30 pounds, the temperature of the water delivered to the boiler will be about 114° F.; if the pressure is 200 pounds, the temperature of the injected water will be about 154° F. The temperature of the water delivered will increase as the delivery of the injector is cut down from its maximum to its minimum. Thus, under 140 pounds steam pressure, and working at its maximum capacity, an injector may deliver water at about 135° F.; while if the injector is cut down to its minimum delivery, the water will be delivered at about 250° F. Under ordinary working conditions, the water is probably delivered at a temperature between 160° and 200° F. The highest temperature at which an injector will lift the feedwater decreases as the steam pressure under which the injector is working is increased. At low steam pressures, the injector may raise water at 125° or 130° F.; while at 140 pounds and upwards, it is not safe to have the water much above 110° F.

16. The number of pounds of water delivered per pound of steam decreases as the steam pressure is increased. At 30 pounds steam pressure, an injector may deliver from 20 to 25 pounds of water per pound of steam; while at 140 pounds pressure, it will deliver only about 13 pounds, and at 180, only about 11 pounds.

17. The term **range**, when applied to an injector, refers to the steam pressure at which it will start and the steam pressure at which it ceases to work. The range of an injector decreases with any increase in the distance that the water must be lifted, and also decreases with any increase in the temperature of the water supply. This is clearly shown in the following table published by the American Injector Company and referring to an injector manufactured by it.

The steam pressure at which injectors of different makes will start varies somewhat, but the range between the starting and stopping pressures with different injectors is practically the same. Most injectors will start on 25 pounds steam pressure, but some are made to start on 15 pounds.

18. Construction.—Injectors may be divided into two general classes—*non-lifting* and *lifting* injectors. They differ from each other, as implied by the name, in that the one class is capable of lifting the water from a level lower than its own, while the other class cannot.

Non-lifting injectors are intended for use where there is a head of water available; consequently, they must be

TABLE I

RANGE OF INJECTORS

Vertical Lift Feet	Feedwater at 60°		Feedwater at 75°		Feedwater at 100°	
	Starting Pressure Pounds per Square Inch	Stopping Pressure Pounds per Square Inch	Starting Pressure Pounds per Square Inch	Stopping Pressure Pounds per Square Inch	Starting Pressure Pounds per Square Inch	Stopping Pressure Pounds per Square Inch
2	15	155	15	145	20	120
4	18	150	18	140		
6	20	142				
8	25	135	25	125		
10	30	125	30	115	35	90
12	35	118				
14	40	110				
15			50	85	45	70
16	45	102				
18	50	90				
20	55	85	55	75		
22	55	75				

placed below the water level. Non-lifting injectors resemble the lifting injector so much in their action that no description of them will be given.

Lifting injectors are of two types—*automatic* and *positive* injectors. Since positive injectors generally have two sets of tubes, they are frequently called *double-tube* injectors.

Automatic injectors are so called from the fact that they will automatically start again in case the jet of water is broken by jarring or other means. They are simpler in construction than double-tube injectors, and they answer very well for a moderate temperature of feed-water supply and not too great a range in steam variation.

Positive, or double-tube, injectors are provided with two sets of tubes, one set of which is used for lifting the water, while the other set forces the water delivered to it by

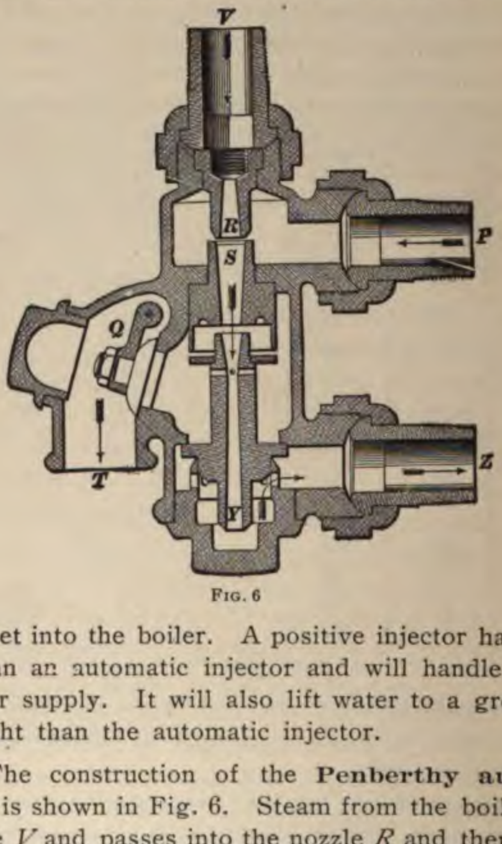


FIG. 6

the first set into the boiler. A positive injector has a wider range than an automatic injector and will handle a hotter feed-water supply. It will also lift water to a greater vertical height than the automatic injector.

19. The construction of the **Penberthy automatic injector** is shown in Fig. 6. Steam from the boiler enters the nipple V and passes into the nozzle R and then into the conical combining tube S. In rushing past the annular opening between R and S, it creates a partial vacuum and causes

water to flow through P, filling the space surrounding the lower end of R and the upper end of S. The nipple P, which is shown at the right-hand side, is really situated in the rear. At first, the mingled steam and water, by reason of the water not having acquired sufficient momentum, do not flow to the boiler; but after the tube Y, the space surrounding it, and the feed-delivery pipe attached to the nipple Z are filled, the mingled steam and water force the swing check-valve Q and pass through the overflow T. As soon as the jet of water passing through the combining tube has acquired sufficient momentum, the boiler check-valve is forced open and the water commences to enter the boiler. In consequence, no more water will enter the space around the lower end of S and the upper end of Y, and there being no pressure in this space, the overflow valve Q will close. The overflow valve is kept closed by the atmospheric pressure on top of it, for, while the injector is working steadily, there will be a partial vacuum in the space around S and Y.

To start the injector, all that is required is to turn on the steam and water. If the steam supply is too great, steam will issue from the overflow; if the water supply is too great, water will issue. Should the jet of water be broken, i. e., fail to enter the boiler, the overflow valve will lift and the mingled water and steam will come out of the overflow until the jet has acquired sufficient momentum to enter the boiler again, when the overflow valve will close for the reasons given.

The automatically closing overflow valve is the distinguishing feature of the automatic injector, and in some form or other is found in all instruments of this class.

20. Fig. 7 is a sectional view of the **Buffalo automatic injector**, which differs considerably in its construction from the Penberthy, but which operates in practically the same manner. This injector needs no valves on the steam and water pipes, the steam-admission valve being controlled by a handle a placed on the valve stem b. With the valve and stem in the position shown, the injector is working. Steam

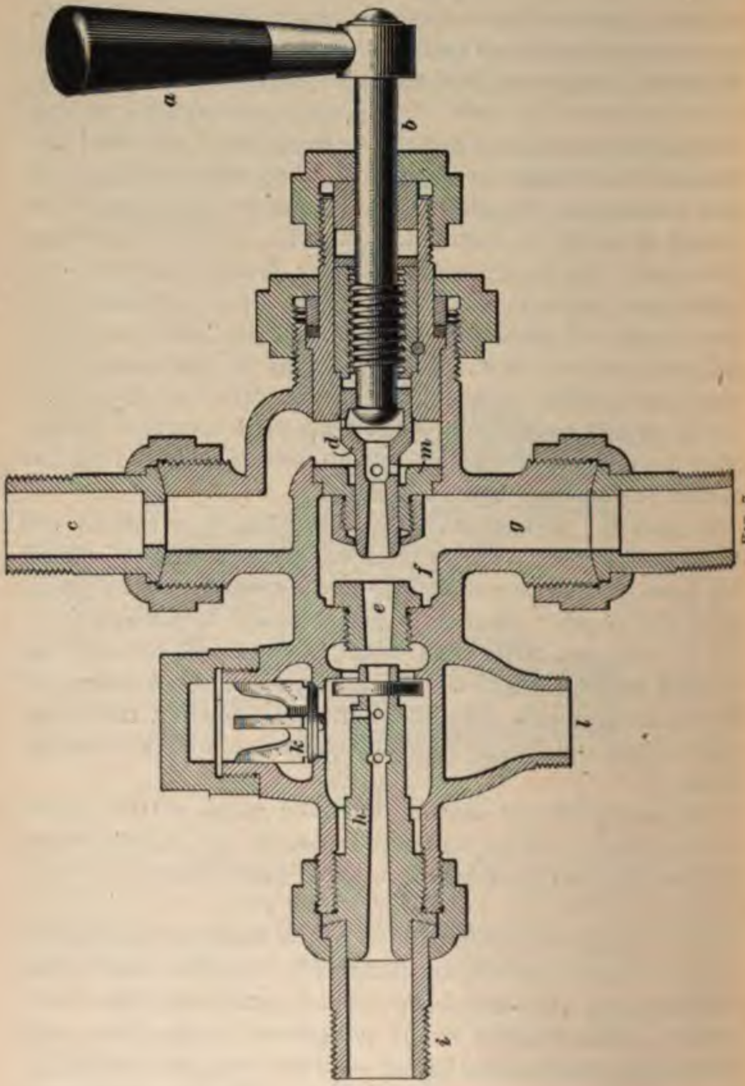


FIG. 7

ɜs through *c* into a chamber surrounding the steam e *d* and through openings in the rear end of the nozzle the latter. In rushing into the suction jet *e*, it carries ɪir in *f* with it, creating a partial vacuum there and ng water to flow through *g*. This water, combined the steam, enters the combining tube *h* and fills the r feedpipe, which is connected to the nipple *i*. At first, et has not sufficient momentum to force the boiler :-valve, and consequently the water flows through the ar opening between *e* and *h* and, after lifting the over-valve *k*, out of the overflow *l*. The speed of the jet ually increases, and as soon as its momentum is suffi- the jet forces the boiler check-valve and enters the r. The overflow valve *k* then closes automatically and ɪjector is working.

the jet should break from any cause, the water will lift verflow valve and come out of the overflow, but as as the momentum is sufficient again, the water will the boiler once more and the overflow valve will close natically.

stop the injector, the handle *a* is turned so as to screw alve stem inwards. The steam nozzle *d*, which is ble longitudinally, remaining at rest, the valve at the f *b* first closes the central opening in the nozzle; then, as otion of the handle continues, the nozzle and valve stem forwards together until the conical seat on the nozzle n the steam-jet guide *m*, when steam is completely shut om the steam nozzle. To start the injector, the valve is slowly turned by means of the handle *a*, which first the central opening of the nozzle, and then, as the e moves backwards with the valve stem, the other open- of the nozzle also admit steam and the injector starts.

. Fig. 8 shows the **Hancock inspirator**, which is f the earliest types of a double-tube injector. The term ator is merely a trade name. Steam from the boiler s through the pipe *a* and flows through the steam e *n* into the combining nozzle *o*, thereby causing water

to flow up the pipe *b* into the lifting side of the instrument. The water then passes in the direction of the arrows to the forcing side of the instrument, entering at the top of the forcing tube *s*, where it is met by a jet of steam flowing through the forcing steam nozzle *r*, and is further heated and given an increased velocity. It then passes through the pipe *c* to the boiler.

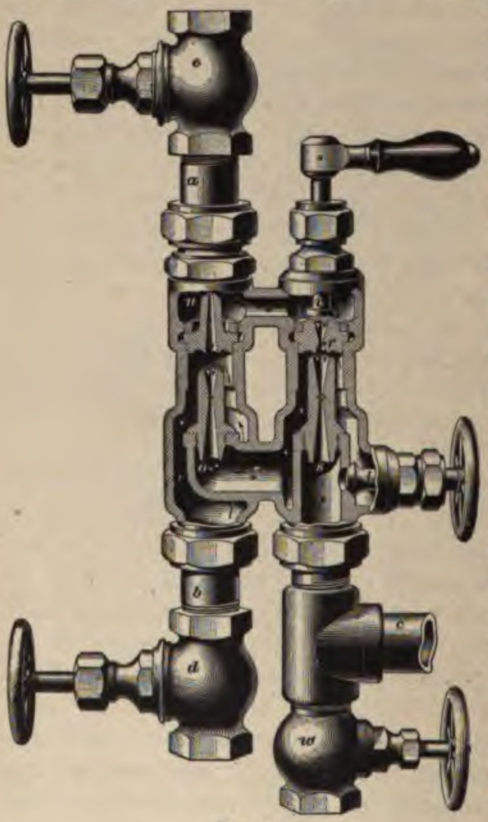


FIG. 8

In order to start the instrument, the valve *v* must be closed and the overflow valves *i*, *w* opened. Next, the water valve *d* and then the steam valve *e* are opened, when the steam will rush through *n*, *o*, *i*, and *w* and out of the overflow until it creates a sufficient vacuum on the left, or lifting, side to cause the water to flow up, which will then discharge out of the overflow. As soon as the water appears, the valve *i* must be closed and the valve *v* opened. Immediately thereafter the overflow valve *w* is to be closed, when the inspirator will be working. To stop the injector, the valves *e* and *v* must be closed and *i* and *w* opened.

22. The **Korting universal double-tube injector** is shown in elevation in Fig. 9 (*a*) and in section in Fig. 9 (*b*).

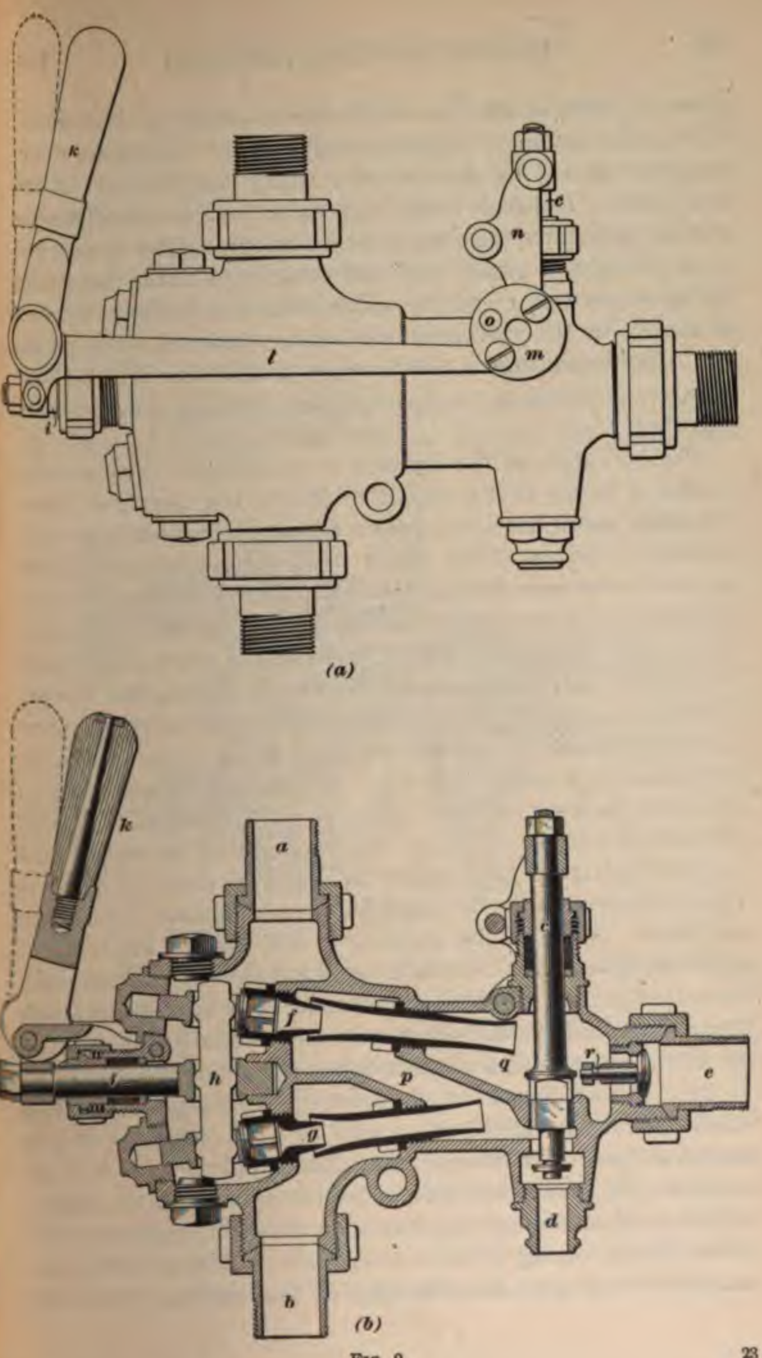


FIG. 9

This injector, in common with the majority of double-tube injectors, contains a mechanically operated overflow valve, which is closed by the act of starting the injector to feed the boiler. In this injector, the lower nozzles constitute the lifting apparatus that delivers the water to the upper nozzles, where it is given sufficient velocity to enter the boiler. Steam enters at *a* and the water enters at *b*; the overflow *d* is closed by the valves on the stem *c*, and the water passes to the boiler through *e*. The steam nozzles *f* and *g* are closed by valves connected by means of the yoke bar *h* to the starting shaft *i*.

The operation of the injector is as follows: The starting handle *k* being in the position shown, the overflow valves are wide open, but the nozzles *f* and *g* are closed by their respective valves. The steam- and water-admission valves are now open and the handle *k* is pulled over gently toward the position shown in dotted lines. This causes the starting shaft *i* and the yoke bar *h* to move in the same direction as the handle, and consequently the valves closing the nozzles *f* and *g* are opened. At the same time, the overflow valves are closed slightly, the starting shaft being connected to the overflow-valve stem by links *l*, bell-cranks *m*, and links *n*. The bell-cranks have their fulcrum at *o*. The steam rushing through the lower nozzle *g* creates a partial vacuum in the water-supply pipe and causes the water to flow up, which is then delivered into the chamber *p* and passes out of the overflow. Some of the water in *p* will pass to the nozzle *f* and will deliver into the chamber *q* and thence into the overflow. In a very short time, the water will be flowing freely from the overflow, and the handle is then pulled over as far as it will go, this operation opening the steam-nozzle valves to their full extent and at the same time closing the overflow openings of the chambers *p* and *q*. The injector is now working, the check-valve *r* being forced open.

Since the overflow outlet is positively closed, the effects of too much steam or water cannot manifest themselves by either fluid coming from the overflow. The effect of too much water will be the stopping of the injector, which can

be told by the absence of vibration in the feedpipe and the comparatively low temperature of the injector. Too much steam manifests itself by the heating of the injector and failure to work; the remedy is either to reduce the steam supply or to increase the water supply, as the heating shows that the steam is not being condensed. These statements apply to all injectors having positively closed overflows, i. e., overflow valves so constructed that they cannot lift automatically when the injector fails to force the water into the boiler.

23. The **Monitor lifting injector,** shown in Fig. 10, occupies an intermediate position between the single-tube

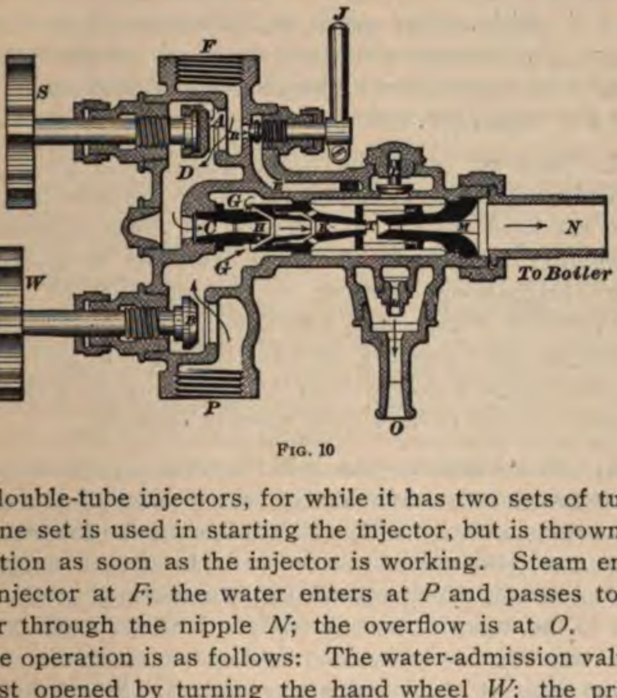


FIG. 10

and double-tube injectors, for while it has two sets of tubes, the one set is used in starting the injector, but is thrown out of action as soon as the injector is working. Steam enters the injector at *F*; the water enters at *P* and passes to the boiler through the nipple *N*; the overflow is at *O*.

The operation is as follows: The water-admission valve *B* is first opened by turning the hand wheel *W*; the primer valve *R* is then opened by the handle *J*, thus permitting steam to flow through the passage *E* and a connection, not shown in the figure, to the nozzle *u*. From *u*, the steam rushes into the overflow nozzle *O*, this nozzle,

in conjunction with the nozzle u, forming the lifting part of a double-tube injector. A passage connects the chamber surrounding u with the space above the overflow valve L. The jet of steam rushing from u through O carries with it some of the air in the chamber to which O is connected, thus forming a partial vacuum in the space above the overflow valve, which opens and thus allows the air in D, C, G, H, K, T, and P to be exhausted. The pressure of the atmosphere now forces the water into the injector, and it finally appears at the overflow. As soon as this happens, the valve R is closed, which throws the priming part of the injector out of action. The steam valve A is now opened by turning the wheel S, which admits steam to the nozzles of the injector proper. At first, the water will come out of the overflow, but as soon as the velocity has become high enough, it will enter the boiler, the overflow valve L closing automatically.

24. Size of Injectors.—The capacity of an injector cannot be calculated by any simple rules; furthermore, there is no standard method followed by all injector makers of designating the capacity of their instruments by numbers or other designations. Hence, an injector must be selected from the lists of capacities published by the makers, selecting one having a delivery per hour at least one-half greater than the evaporation per hour of the boiler to which it is applied, in order to have some reserve capacity.

25. Installation.—An injector must always be placed in the position recommended by the maker, for the reason that some injectors will work well only in one position. There must always be a stop-valve in the steam-supply pipe to the injector, which should, for convenience, be placed as close to the injector as is possible. While lifting injectors, when working as such, scarcely need a stop-valve in the suction pipe, it is advisable to supply it. When the water flows to the injector under pressure, a stop-valve in the water-supply pipe is a necessity. A stop-valve and check-valve must be placed in the feed-delivery pipe, with the stop-valve next to the boiler. The check-valve should never

be omitted, even though the injector itself is supplied with one. No valve should ever be placed in the overflow pipe, nor should the overflow be connected directly to the overflow pipe, but a funnel should be placed on the latter so that the water can be seen. This direction does not apply to the inspirator or to any other injector that has a hand-operated, separate overflow valve. In the inspirator, the overflow pipe is connected directly to the overflow, but the end of the pipe must be open to the air. In general, where the injector lifts water, it is not advisable to have a foot-valve in the suction pipe, as it is desirable that the injector and pipe drain themselves when not in use. It is a good idea to place a strainer on the end of the suction pipe.

The steam for the injector must be taken from the highest part of the boiler, as it is essential to the successful working of the injector that it be supplied with dry steam. Under no consideration should the steam be taken from another steam pipe; the injector should always have its own independent steam-supply pipe. The suction pipe should be as straight as possible and must be absolutely air-tight. A very important consideration in connecting up an injector is to have the pipes cleaned by blowing them out with steam before making the connection, since quite a small bit of dirt getting into the injector will interfere seriously with its working. It is recommended to always so locate the injector that the steam pipe, suction pipe, and feed-delivery pipe will be as straight and as short as possible. With boilers that are forced very much and hence generate wet steam, it is advisable to use a so-called *supplementary dome*, which is simply a vertical piece of, say, 2-inch pipe about 12 to 18 inches long; the injector steam pipe is then connected to the top of this supplementary dome.

26. Troubles and Remedies.—In discussing the difficulties experienced with injectors, the suction pipe, strainer, feed-delivery pipe, and check-valve are considered as parts of the injector, since a disorder in any of these affects the work of the injector itself. In searching for the cause of a trouble,

therefore, the suction and delivery pipes should be carefully inspected, as well as the injector.

27. The causes that prevent an injector raising water are:

1. *Suction Pipe Stopped Up.*—This is due, generally, to a clogged strainer or to the pipe itself being stopped up at some point. This prevents water from coming through and is probably the most frequent cause of an injector not priming. In case the suction pipe is clogged, blow steam back through the pipe to force out the obstruction.

2. *Leaks in Suction Pipe.*—When this is the case, air enters and prevents the injector forming the vacuum required to raise the water. To test the suction pipe for air leaks, plug the end and turn the full steam pressure on the pipe. The presence of leaks will be revealed by the steam issuing therefrom. To get steam into the suction pipe, the overflow valve must be held to its seat in some convenient way, or the overflow must be closed tight in some manner. It is advisable to have the suction pipe full of water before steam is turned on, since the presence of small leaks will be revealed better by water than by steam. After testing, do not forget to clear the overflow and the end of the suction pipe.

3. *Water in the Suction Pipe Too Hot.*—In case the feed-water supply is taken from a tank and the supply is cool under normal conditions, a leaky steam valve or leaky boiler check-valve and leaky injector check-valve may be the cause of hot water or steam entering the source of supply and heating the water so hot that the injector refuses to handle it.

The reason that hot water in the suction pipe affects the operation of the injector is as follows: The temperature at which water boils depends on the pressure to which it is subjected. It has been determined, by experiment, that water will boil at about 380° F. under a gauge pressure of 180 pounds; at 212° F. when subjected to an atmospheric pressure of about 14.7 pounds per square inch; and at about 190° F. when in 10 inches of vacuum. This shows that decreasing the pressure on the water lowers its boiling point.

Now, when the lifting jet of an injector is turned on, a vacuum is formed in the suction pipe; and if the water there is at a temperature of 160° to 175° F., it gives off vapor, which fills the suction pipe and destroys the vacuum.

In case the water is too hot, cool it in any convenient manner and at the first opportunity trace out the cause and remove it. When the water in the suction pipe is very hot, but the water in the source of supply is cool, hold the overflow valve to its seat in any convenient manner or plug the overflow and open the steam valve. The steam pressure will then force the hot water out of the suction pipe. Open the overflow valve or overflow as soon as this has been done; the cool water entering the suction pipe should now be raised easily. Hot water in the suction pipe only is most likely due to a choked overflow.

4. *Obstruction in Tubes.*—There may be an obstruction in the lifting or combining tubes, or the spills (or openings) in the tubes through which the steam and water escape to the overflow may be clogged up. In either case, the free passage of the steam to the overflow will be interfered with, and, consequently, a steam pressure instead of a vacuum will be formed in the suction pipe, the extent of the pressure depending on the amount of obstruction.

28. In some cases an injector will lift water, but will not force it into the boiler; or it may force part of it into the boiler and the rest out of the overflow. When it fails to force, the trouble may be due to one or the other of the following causes:

1. *Choked Suction Pipe or Strainer.*—If the suction pipe or the strainer is partially choked, the injector, in either case, will be prevented from lifting sufficient water to condense all the steam issuing from the steam valve. The uncondensed steam, therefore, will gradually decrease the vacuum in the combining tube until it is reduced so much that the injector will break. It is to be remembered that when the injector is operating, it is the vacuum in the combining tube that causes the water to be raised. The remedy,

in case the supply valve is partially closed, is obvious. In the case of choked suction pipe, blow out the obstruction.

2. *Suction Pipe Leaking.*—The leak may not be sufficient to entirely prevent the injector lifting water, but the quantity lifted may be insufficient to condense all the steam, which, therefore, destroys the vacuum in the combining tube. A slight leak may exist that will simply cut down the capacity of the injector. In such a case, an automatic injector will work noisily, on account of the overflow valve seating and unseating itself as the pressure in the combining tube varies, due to the leak.

3. *Boiler Check-Valve Stuck Shut.*—If completely closed, the injector may or may not continue to raise water and force it out of the overflow: it depends on the design of the injector. If the boiler check is partly open, the injector will force some of the water into the boiler and the remainder out of the overflow. In case the check-valve cannot be opened wide, water may be saved by throttling both steam and water until the overflow diminishes, or, if possible, ceases. The steam should be throttled at the valve in the boiler steam connection, however, and not at the steam valve of the injector, as throttling tends to superheat the steam, and an injector will not work as satisfactorily with superheated steam as with saturated steam. By throttling the steam at the boiler, the excess of heat due to this throttling will be lost before the steam reaches the injector.

If a check-valve sticks, it can sometimes be made to work again by tapping lightly on the cap or on the bottom of the valve case.

4. *Obstruction in Delivery Tube.*—Any obstruction in the delivery tube, such as cotton waste, scale, or coal, will cause a heavy waste of water from the overflow. To remedy this, the tube will probably have to be removed and cleaned.

5. *Leaky Overflow Valve.*—This not only diminishes the capacity of the injector, but allows air to be drawn into the boiler; and if the leak is sufficiently great, it will destroy the vacuum in the combining tube and prevent the injector operating. A leaky overflow valve is indicated by the boiler

heck chattering on its seat. To remedy this defect, grind he valve on its seat until it forms a tight joint.

6. *Injector Choked With Lime.*—It is essential to the proper working of an injector that the interior of the tubes should be perfectly smooth and of the proper bore. When the injector handles water containing scale-forming impurities, such as are found in river and lake waters, which are chiefly sulphate of lime and carbonate of lime, the tubes, in course of time, become covered with a deposit of lime and the capacity of the injector decreases until, finally, it refuses to work at all. If the water used is very bad, it becomes necessary to frequently cleanse the tubes of the accumulated lime. This may be accomplished by putting the parts in an acid bath, allowing the acid to remove the scale. The bath should consist of one part of muriatic acid to ten parts of water. The tubes should be removed from it as soon as the gas bubbles cease to be given off. The acid combines with the lime and forms a gas, and as long as there is lime to combine with, it will not attack the copper in the tubes. After the lime has all combined, however, the acid will attack the tubes, with the result that the inner surface will become pitted and rough, which will affect the working of the injector.

29. Advantages and Disadvantages.—The advantages of the injector as a boiler-feeding apparatus are its cheapness as compared with a pump of equal capacity; it occupies but little space; the repair bills are low, owing to the absence of moving parts; no exhaust piping is required, as with a steam pump; it delivers hot water to the boiler. The disadvantages of the injector are that it will not start with a steam pressure less than that for which it is designed, and that it will stand but little abuse, being poorly adapted for handling water containing grit or other matter liable to cut the nozzles.

Such economy as is derived from the use of an injector is not chargeable to its economy in the use of steam, for it uses as much steam as a fairly good steam pump, but rather

to the fact of the use of an injector insuring the delivery of hot feedwater to the boiler. The introduction of hot feedwater has a marked effect on the repair bills and tends to increase the life of a boiler by diminishing the stresses incidental to expansion and contraction.

FEEDWATER PURIFICATION AND HEATING

FEEDWATER PURIFICATION

IMPURITIES

30. The solid matter contained in sea-water consists chiefly of chloride of sodium (common salt), sulphate of lime (plaster of Paris), carbonate of lime (limestone or marble), chloride of magnesium, and traces of various other substances. River and lake waters, if fresh, as is usually the case, are liable to contain carbonate of lime, sulphate of lime, carbonate of magnesia, sulphate of magnesia, organic matter, earthy matter, and occasionally acids. Fresh water from a surface condenser or the water drawn from a jet condenser usually contains the oil or grease used for lubricating the engine cylinders, which is carried into the condenser by the exhaust steam.

Carbonate of lime will not dissolve in pure water, but will dissolve in water that contains carbonic-acid gas. It becomes insoluble and is precipitated in the solid form when the water is heated to about 212°, the carbonic-acid gas being driven off by the heat.

Sulphate of lime dissolves readily in cold water, but not in hot water. It precipitates in the solid form when the water is heated to about 290°, corresponding to a gauge pressure of 45 pounds.

Chloride of sodium will not be precipitated by the action of heat unless the water contains large quantities of it. Since it

is generally present in but very small quantities in fresh water, it will take a very long time before it causes trouble in the boiler; and if the boiler is blown out at the usual intervals, there will be little danger of the water becoming saturated with it. Consequently, it is one of the least troublesome scale-forming substances contained in fresh water. In sea-water, however, it is present in such a large proportion that the use of sea-water for boiler feeding, except in emergencies, is precluded.

Chloride of magnesium is one of the worst impurities in water intended for boilers, for while not dangerous as long as the water is cold, it renders the water corrosive when heated, and dangerously corrosive when it is present in large quantities; in such a case the metal of the boiler is rapidly corroded.

Carbonate of magnesia and *sulphate of magnesia* are not very troublesome constituents of feedwater.

Organic matter by itself may or may not cause the water to become corrosive, but will often cause foaming; when it is present in small quantities in water containing carbonate or sulphate of lime, or both, it usually serves to keep the deposits from becoming hard.

Earthy matter, like organic matter, is not dissolved in the water, but is in mechanical suspension. It is very objectionable, especially in the form of clay; and when other scale-forming substances are present, a hard scale resembling Portland cement is likely to result.

Acids, such as sulphuric acid, nitric acid, tannic acid, and acetic acid, are sometimes present in feedwater taken from rivers. The sulphuric acid is the most dangerous of these acids, attacking the metal of which the boiler is composed and corroding it very rapidly. The other acids, while not so violent in their action as the sulphuric acid, are also dangerous, and water containing them should be neutralized when it must be used.

Oil, or *grease*, in the feedwater, when carried into the boiler, is liable to do much harm; it seems to combine mechanically with certain impurities, forming a loose, spongy mass, or

deposits by itself, on the plates. In either case, it greatly hinders the transfer of heat from the plate to the water, and hence has been in numerous instances the cause of overheated plates and collapsed furnace flues. Too great care cannot be exercised to keep oil or grease out of boilers.

METHODS OF PURIFICATION

31. The impurities contained in the feedwater of marine boilers may be removed or rendered harmless in several ways:

1. *By Filtration.*—This method will remove floating impurities, such as oil or grease mixed with the feedwater of a condensing engine. It will also quite effectually remove all matter in mechanical suspension, such as earthy matter. Filtration pure and simple will not remove matter in solution.

2. *By Gravity Separation.*—This method relies on the difference in specific gravity between oil or grease and water for a separation. It will not separate matter in solution from the feedwater.

3. *By Heat.*—This method will precipitate carbonate of lime, sulphate of lime, and chloride of sodium, the three scale-forming substances held in solution. The carbonate of lime and the sulphate of lime precipitate as soon as the water is heated to about 290° F. Hence, if the feedwater be heated in a separate vessel to that temperature, the impurities will deposit there instead of in the boiler. Chloride of sodium (salt) will not be precipitated by heating the water, unless the water is saturated with it. Chloride of magnesium cannot be removed from the water by heating.

4. *By Chemical Means.*—This method will render harmless the choloride of magnesium contained in solution in sea-water. When water containing chloride of magnesium in a proportion of more than about 200 grains to the gallon is heated to a high temperature, the water will, under certain conditions, particularly if corrosion has already begun in the boiler, become acid, and hence highly corrosive. Chemical means will also render harmless fatty acids due to vegetable

or animal oils, or adulterated mineral oils that have been decomposed by heat. In marine work, it is extremely rare to attempt to purify fresh feedwater by chemical means.

PURIFYING BY FILTRATION

32. A cheap, easily installed, and quite efficient way of removing oil or grease and other floating impurities from feedwater is to pass the water through an open wooden box filled with hay. The suction pipe of the feed-pump is connected to the bottom of this box, the end of the suction pipe being covered with a perforated plate called a *strainer*, to prevent any hay from entering the pump. The oil and grease contained in the water will deposit on the hay, which is renewed occasionally. The box is usually divided by wooden partitions into several compartments, the passage of the water being as follows: It enters at the bottom of one compartment, flows upwards through the hay therein, and then enters the top of the next compartment through a hole near the top, or over the top, of the partition. It then flows downwards into the second compartment and enters the third at the bottom, and so on, leaving finally at the bottom. Sometimes burlap (ordinary coarse bagging) is placed in the first compartment for the water to percolate through. In doing so, it deposits most of the impurities on the burlap, which must be cleaned and renewed frequently.

33. A **Ross feedwater filter,** designed to remove oil and grease from the feedwater of a surface condensing engine, is shown in Fig. 11. The water coming from the feed-pump enters at *a* and passes into the filtering chamber *b*. It cannot leave this filtering chamber without passing through the filter *c*, which consists of light circular bronze sections of open latticework held together by long bolts and covered by toweling. This material is technically known as "linen terry," and popularly as "Turkish toweling." The toweling is made up in the shape of a bag somewhat larger than the spider; it is drawn over the filter and down between each of the sections by a string wound around it. The

feedwater slowly passes through the filtering material into the interior of the filter; it then goes through the left-hand opening of the filtering chamber and through the valve *d* into the feedpipe again, and thence to the boiler. The foreign matter filtered from the water accumulates on the filtering material, and in course of time offers considerable resistance to the passage of the water. This resistance is shown by the difference in reading of two pressure gauges. One of these is connected to the chamber *b* and the other to the left-hand passage. When this difference amounts to 3 pounds, the filter is in need of cleaning. To clean the filter, close valves *d*, *e* and open the drain at *f*. Now open valve *e* a little. A current of water will then flow around the filter and out of the drain, washing the outside of the toweling. Next, close valve *e* and open valve *d*. Then, the drain being open, a current of water will flow through the filter in a direction opposite to that in which the water passes through it when filtering. The water, flowing in a reverse direction, tends to loosen the foreign matter adhering to the outside of the filter. To start the filter again, open valves *d* and *e* and close the drain. Should it be found that the washing of the filter as explained above is insufficient to clean it, new toweling must be inserted. To do this, close valves *d* and *e* and open the drain. The water from the feed-pump will now pass directly to the boiler, the screwing down of the valve *e* to close the opening to the filter chamber opening a by-pass, as shown. The cover *g* can now be removed and new toweling inserted.

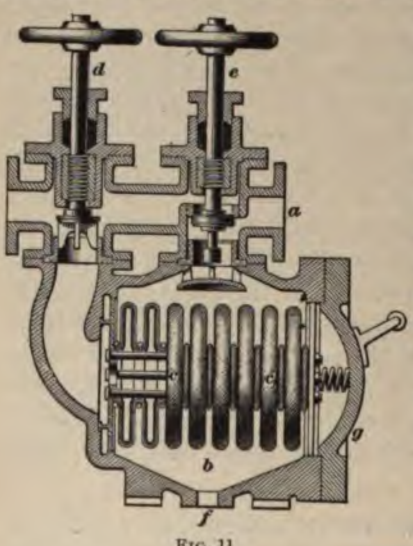


FIG. 11

34. An **Edmiston feedwater filter** is shown in Fig. 12. It consists of a vessel *a* divided into two chambers by perforated plates *b*, *b* covered with coarsely woven cloth. The feedwater is admitted to the chamber *a'*, and cannot reach the chamber *a''* except by passing through the filtering cloth. The oil and other floating impurities rise into the scum chamber *c*, whence they are removed, periodically, by opening the blow-off *d*. The heavier impurities settle into the pocket *e*,

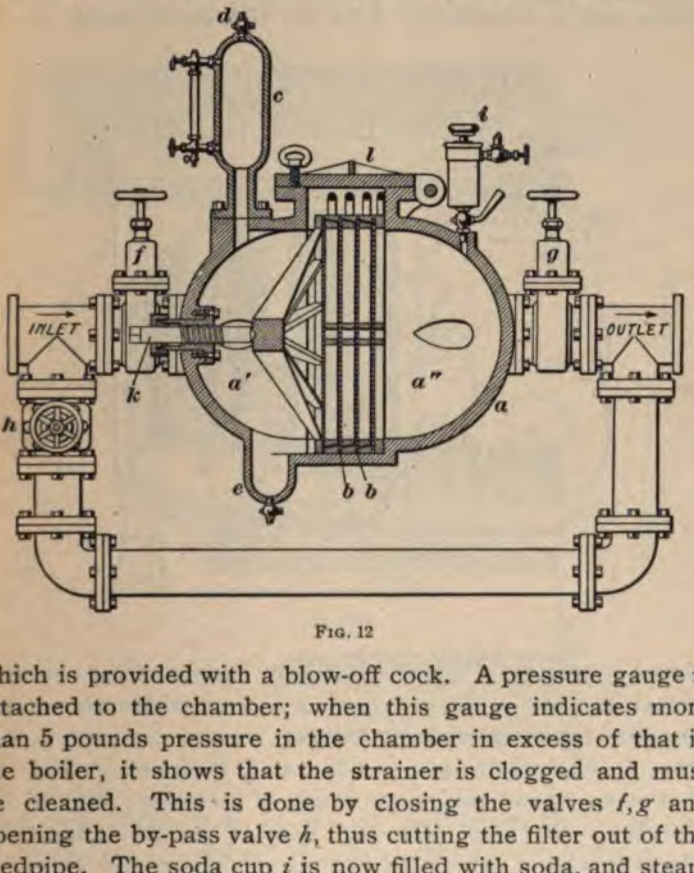


FIG. 12

which is provided with a blow-off cock. A pressure gauge is attached to the chamber; when this gauge indicates more than 5 pounds pressure in the chamber in excess of that in the boiler, it shows that the strainer is clogged and must be cleaned. This is done by closing the valves *f*, *g* and opening the by-pass valve *h*, thus cutting the filter out of the feedpipe. The soda cup *i* is now filled with soda, and steam turned on, thus boiling out the filter. The soda dissolves the grease and the matter in the filter can be blown out.

If boiling out fails to clear the filter, the filtering cloths must be removed and new ones substituted. To do this, first cut the filter out of the feedpipe, letting the feedwater go through the by-pass. Then loosen the setscrew *k*. Now open the hinged door *l*. The plates or diaphragms can then be readily removed.

35. The feedwater filter that is illustrated in Figs. 13, 14, and 15 is known to the trade as the **Reflex feedwater filter** and is manufactured by the Blackburn-Smith Co., of

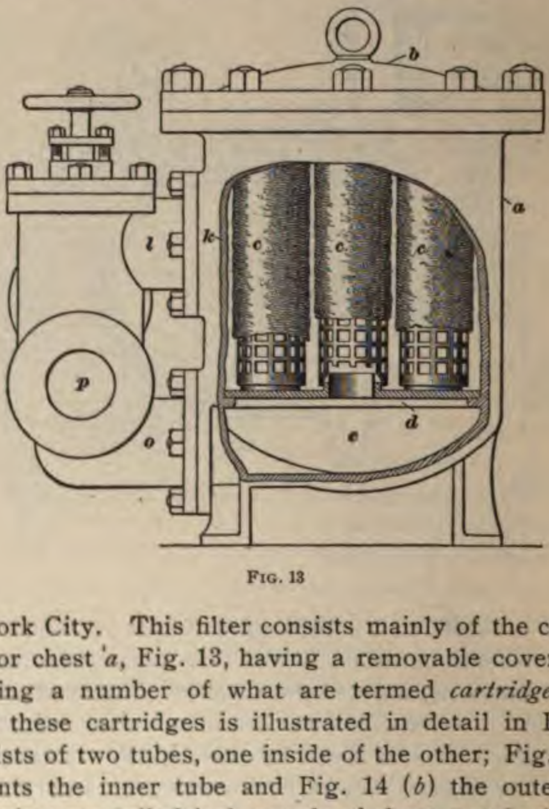


FIG. 13

New York City. This filter consists mainly of the cast-iron vessel or chest *a*, Fig. 13, having a removable cover *b*, and containing a number of what are termed *cartridges* *c, c, c*. One of these cartridges is illustrated in detail in Fig. 14. It consists of two tubes, one inside of the other; Fig. 14 (*a*) represents the inner tube and Fig. 14 (*b*) the outer tube. These tubes are cylindrical vessels of sheet metal, closed at their tops and perforated with large rectangular holes; in fact, they are merely light frames for the support of the

filtering material. The lower end of the inner tube is permanently expanded in the plate *d*, Figs. 13 and 14 (*a*), which separates the filter chest from the pure-water chamber *e*.

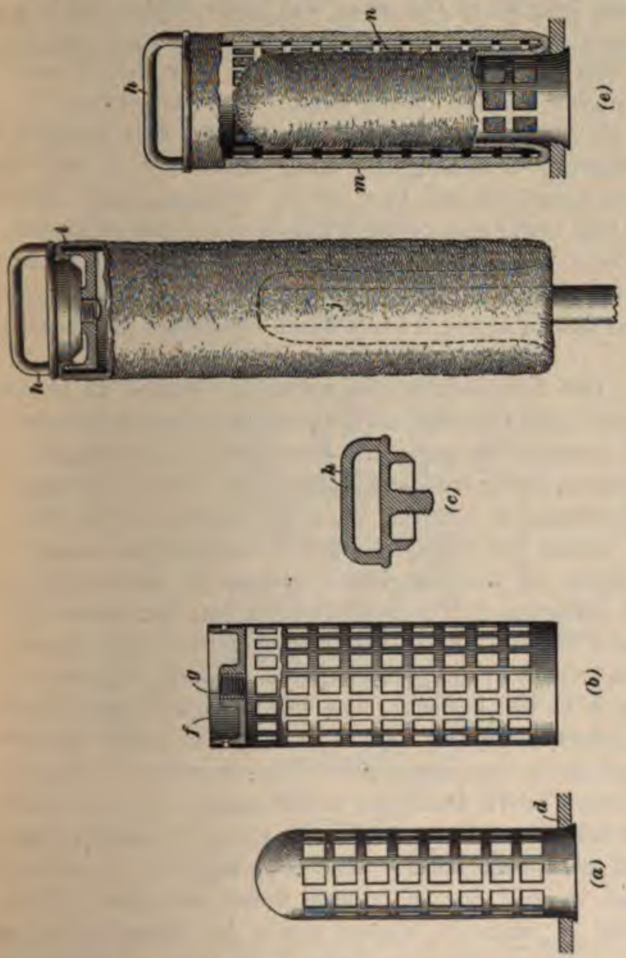


FIG. 14

The outer tube, Fig. 14 (*b*), is removable; its upper end is closed by the flanged stopper *f*, which is riveted in. A tapped hole *g* is provided in the center of the stopper to receive the threaded part of the handle and cloth cramp *h*,

Fig. 14 (*c*). The filtering material consists of a bag made of linen or cotton special Turkish toweling, left open at the top end. The length of the bag is a little more than the combined lengths of the inner and outer tubes, and it is of the proper size to fit snugly over the outside of the outer tube. The bag is drawn over the outer tube until it extends ½ inch above the top of the tube. It is then turned in as shown at *i*, Fig. 14 (*d*), and the handle *h* is screwed in place, thereby cramping the cloth effectually at that end. The loose end of the bag, which is closed and extends beyond the lower end of the outer tube, is shoved inside, as shown at *j*, Fig. 14 (*d*), by any convenient rod or stick. The outer tube is then placed over the inner tube telescopically, as shown in Fig. 14 (*e*), which completes the cartridge.

36. The filtration of the water is effected as follows: The water from the feed-pump enters the filtering chamber *k*, Fig. 13, through the pipe *l* and surrounds the cartridges *c*, *c*, *c*. The water is under sufficient pressure to force it through the outer covering of filtering cloth shown at *m*, Fig. 14 (*e*). It now enters the annular space *n* between the inner and outer tubes of the cartridge, whence it passes through another thickness of the filtering cloth into the inside of the inner tube and out through the lower end of it into the pure-water chamber *e*, Fig. 13, and thence through the pipe *o* and opening *p* to the boilers. The lower parts of the filtering cloths are cut away in Fig. 13 in order to partly show the construction of the cartridges. The number of cartridges varies from 1 to 19, according to the capacity of the filter, and a spare set of bags, which should always be kept clean, is supplied with each filter. The foul bags can be removed and then washed out in boiling water and soda. Thus a clean set of bags is always ready for insertion at any moment. The advantages claimed for these cartridges are simplicity, ease of removal, large filtering surface in a small space, effectiveness due to double filtration, saving of weight, and the rapidity with which the filtering bags can be taken out and renewed.

37. The valves and other details of the Reflex filter are illustrated in Fig. 15. The inlet, outlet, and by-pass valves are operated by a single stem *a*. While the filter is in use, the valve stem is screwed up as far as it will go. This opens the inlet valve *b* and the outlet valve *c* simultaneously.

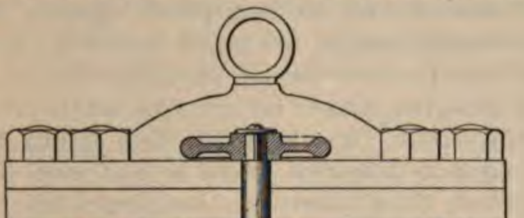


FIG. 15

At the same time, the upper edge *d* of the outlet valve closes the by-pass passage. To clean the filter, the valve stem is screwed down hard, thereby closing the inlet and outlet valves and opening the by-pass valve. This allows the boilers to steadily receive their normal amount of feedwater. Aside from the fact that with a single stem weight is saved

and simplicity is gained, the most important feature of this arrangement is that it is impossible to fracture the feed-pump or feedpipe by closing the inlet valve before opening the by-pass valve, which might be the case if three distinct valves were used.

38. Filters of over 150-horsepower capacity are fitted with two pressure gauges, one on the inlet side *e* and one on the outlet side *f* of the chest. The difference of pressure shown by these two gauges indicates the actual pressure on the filtering cloths. To obtain the best results from the filter, the pressure difference should be kept under 30 pounds per square inch; if the difference of the gauges indicates more than this, the filter is in need of cleaning. The drain and sludge valve *g* is placed at the lowest point of the filter chest, through which the oil and dirt can be blown out. The steam-cleaning valve *h*, which is attached to the pure-water chamber, is provided to temporarily clean the filter without removing the cartridges. This is effected by by-passing the feedwater and then opening the steam-cleaning and drain valves. The steam flows in the reverse direction to the flow of water and blows a large portion of the grease and sediment through the drain valve.

39. When it is desirable to provide for constant filtration in preference to by-passing the greasy water into the boilers while the filter is being cleaned, the duplex filter meets the requirement. This consists simply of a pair of duplicate filters connected up by means of the necessary piping and ordinary cross and angle stop-valves, so arranged that either filter can be entirely shut off for cleaning, while the other, for the time being, does the work of both. These filters are tested to 300 pounds hydrostatic pressure per square inch.

PURIFYING BY GRAVITY SEPARATION

40. A **Wass grease extractor**, as made by Cramp & Co., is illustrated in Fig. 16. It consists of a closed cast-iron vessel placed between the feed-pump and the boiler. The feedwater enters at *A* and passes over and under the

alternate partitions B, B, leaving the extractor at C. The grease, on account of its lower specific gravity, rises to the surface in each of the compartments formed by the partitions, and since the level of the water is above the top of the higher partitions, the tendency of the grease on the surface is to flow toward the outlet end of the apparatus, where the grease outlet pipe D is located. The pressure exerted by the air confined between the surface of the liquid and the

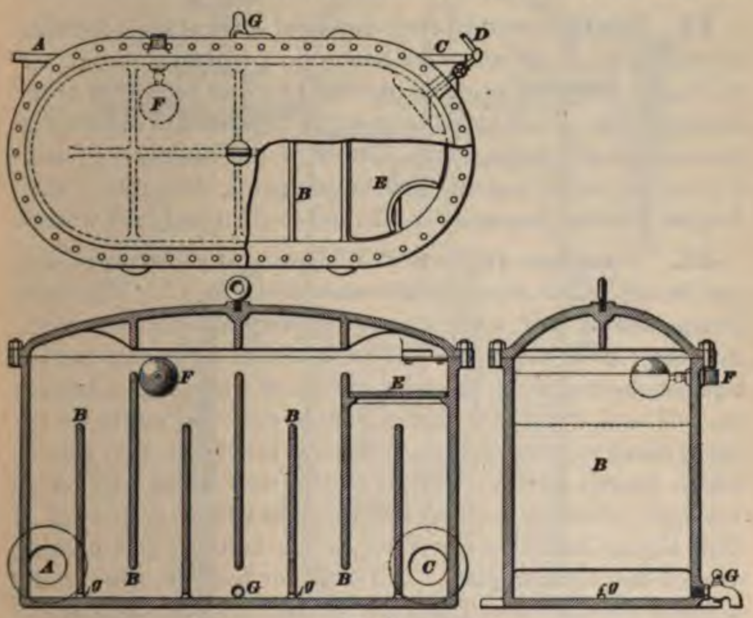


FIG. 16

cover forces out the grease on opening the valve in the pipe D. This is done at intervals varying with the amount of grease carried in. A plate E at the outlet end prevents the mixture of the outflowing grease and feedwater. A float valve F allows the air to escape from the extractor when it is first filled. The valve is closed by means of the float, on the water reaching a certain height, and the air remaining is compressed. By means of the drain-cock G, the extractor may be emptied for the purpose of examination or repair,

the different compartments communicating with each other by a small hole *g* at the bottom of each partition. This style of extractor is extensively used in sea-going steam vessels having surface condensers. The pipe connections are made so that the extractor may be cut out, if desired, and the feedwater passed directly to the boiler.

PURIFYING BY HEAT

41. Heat causes the precipitation of several scale-forming substances. If the water be heated in a separate vessel, and if a large, quiet chamber be provided for the reception of the heated water, a considerable quantity of matter in mechanical suspension will settle at the bottom of the chamber, in addition to the matter precipitated by becoming insoluble. Purification by heat is especially adapted to river and lake waters.

42. A **Buffalo feedwater heater and purifier,** which can be applied to any boiler, is shown in Fig. 17. The feed-pumps deliver their water through the pipe *e* and check-valve *f* into the top of the heater. The entering feedwater strikes against the top head; the solid stream of water is thus broken up. It now flows in a zigzag course over the edges of the spray disks *g*, *g*, being thus spread out into large, thin sheets, which readily absorb the heat of the live steam with which the spray chamber is filled and which is admitted through *d*. The highly heated water falls to the bottom, and passing around the division plate *l* and deflector plate *m*, rises in the settling chamber *a*. Thence it passes into the feedpipe *c*. The feedwater being heated to almost the same temperature as that in the boiler, the scale-forming substances precipitate and collect at the bottom of the settling chamber, whence they can be removed by opening the blow-off valve *j*. Nearly all foreign matter in mechanical suspension also collects in this settling chamber, and is removed with the scale-forming substances. The impurities that have a smaller specific gravity than water rise to the top of the settling chamber and float on the water. By extending the equalizing pipe *i*, which forms the feed-outlet from the heater,

below the surface of the water, the floating impurities are prevented from entering the feed-outlet. The heater should be placed above the boilers; the water will then flow into the boilers by gravity. To prevent the water in the boilers from backing up into the heater when the blow-off of the heater is opened, an automatic shut-off valve, which is simply a

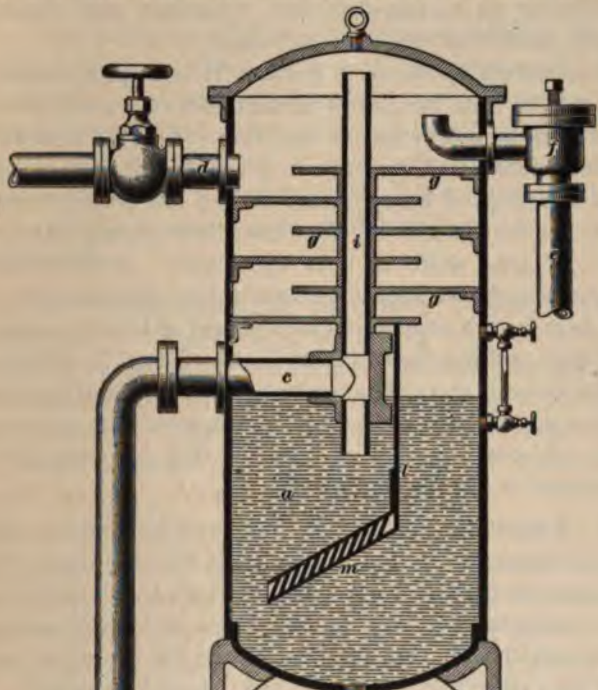


FIG. 17

special form of check-valve, is supplied. This valve is placed in the feedpipe between the heater and the boilers. Under ordinary working conditions, it is sufficient to blow out the heater once every 6 hours. Siphonage through *c* is prevented by the equalizing tube *i*, which is open on top, and insures equal pressures inside of *c* and the heater at all times.

PURIFYING BY CHEMICAL MEANS

43. When sea-water is mixed with the feedwater of surface condensing engines, the corrosive effect of the chloride of magnesium thus admitted can be neutralized by using the ordinary unslaked lime of commerce. The lime converts the chloride of magnesium into magnesia and chloride of calcium, neither of which is corrosive.

When starting with new boilers, it is recommended that 10 pounds of lime be placed in the boilers for every thousand indicated horsepower on the first day. For the next 6 days' continuous steaming, use about 5 pounds of lime for every thousand indicated horsepower. When the boilers are examined at the expiration of this time, they should have a thin coating of lime scale all over the inside. If this is not the case, the use of lime should be continued. Mix the finely powdered lime in the proportion of 1 pound of lime to a gallon of water, thus making the so-called "milk of lime." Introduce it into the hotwell in any convenient manner in small quantities.

It is a good plan to use lime continuously to prevent corrosion. About 1 pound of lime per day for each thousand horsepower is usually sufficient.

44. Feedwater taken from rivers and lakes often contains much carbonate of lime or sulphate of lime, or both. Water containing carbonate of lime may be treated with caustic soda, which precipitates the carbonate of lime and leaves carbonate of soda, which is harmless. For every 100 grains of carbonate of lime 80 grains of caustic soda should be added.

45. Sal ammoniac is sometimes added to water containing carbonate of lime and will cause the latter to precipitate. Its use is not advisable, however, on account of the danger of the formation of hydrochloric acid, which will attack the boiler. The formation of this acid is due to an excessive quantity of sal ammoniac having been used.

46. While slaked lime will precipitate carbonate of lime, it will have no effect on sulphate of lime, and water containing the latter, either alone or in conjunction with

carbonate of lime, must be treated with other chemicals. The most available ones for water containing both are carbonate of soda and caustic soda. These are often fed into the boiler and will precipitate the carbonate of lime and sulphate of lime there, requiring the sediment to be blown out or otherwise removed periodically.

47. The action of caustic soda on carbonate of lime and sulphate of lime in water containing both these ingredients is as follows: The soda precipitates the carbonate of lime, and in so doing carbonate of soda is formed, which, in turn, combines with the sulphate of lime, precipitating it in the form of carbonate of lime, and in so doing forming sulphate of soda, which is very soluble and harmless and may long be neglected.

48. When treating water containing carbonate of lime and sulphate of lime, caustic soda may be used either by itself or in combination with carbonate of soda, depending on the relative proportions of carbonate of lime and sulphate of lime present in the water. The amount of caustic soda or carbonate of soda to be used per gallon of feedwater, of 231 cubic inches, can be found as follows:

Rule I.—*Multiply the number of grains of carbonate of lime per gallon by 1.36. If this product is greater than the number of grains of sulphate of lime per gallon, only caustic soda is to be used. To find the quantity of caustic soda required per gallon, multiply the number of grains of carbonate of lime in a gallon by .8.*

Rule II.—*Multiply the number of grains of carbonate of lime per gallon by 1.36. If this product is less than the number of grains of sulphate of lime per gallon, take the difference and multiply it by .78 to obtain the number of grains of carbonate of soda required per gallon. To find the amount of caustic soda required per gallon, multiply the number of grains of carbonate of lime in a gallon by .8.*

EXAMPLE.—A quantitative analysis of a certain feedwater shows it to contain 23 grains of sulphate of lime and 14 grains of carbonate of lime per gallon; how much caustic soda and carbonate of soda should be used per gallon to precipitate the scale-forming substances?

SOLUTION.—By rule I, 14 × 1.36 = 19 gr. Since this product is less than the number of grains of sulphate of lime per gallon, rule II is to be used. Applying rule II, it is found that (23 − 19) × .78 = 3.12 gr. of carbonate of soda, and 14 × .8 = 11.2 gr. of caustic soda are required. Ans.

49. Water containing sulphate of lime, but no carbonate of lime, may be treated with carbonate of soda. The amount of the latter that is required per gallon to precipitate the sulphate of lime is found by multiplying the number of grains per gallon by .78.

50. When using soda, it is well to keep in mind that it will not remove deposited lime from the inside of a boiler. All that the soda can do is to facilitate the separating of the lime; i. e., cause it to deposit in a soft state. This sediment must be removed periodically.

51. For decomposing sulphate of lime, tribasic sodium phosphate, more commonly known as trisodium phosphate, is often used. This is claimed to act on the sulphate of lime, forming sulphate of sodium and phosphate of lime, the former of which remains soluble and is harmless, while the latter is a loose, easily removed deposit. Trisodium phosphate also acts on carbonate of lime and carbonate of magnesia, forming phosphate of lime and phosphate of magnesia, at the same time neutralizing the carbonic acid released from the carbonate of lime and magnesia, and the sulphuric acid released from the sulphates.

52. Acid water can be neutralized by means of an alkali, soda probably being the best one. The amount of soda to be used can best be found by trial, adding soda until the water will turn red litmus paper blue.

TESTING WATER

53. A quantitative analysis of feedwater can be made only by an expert chemist having a well-appointed laboratory and the proper apparatus; a qualitative analysis for the most common impurities can be easily made, however, with the aid of chemicals procurable in almost any drug store. Such

an analysis will show what kinds of impurities are present, but will not show the amounts.

It is a good plan to test the feedwater and also the water in the boiler occasionally for corrosiveness. This may be done by placing a small quantity in a glass tumbler and adding a few drops of methyl orange. If the sample of water is acid, and hence corrosive, it will turn pink. If it is alkaline, and hence harmless, it will be yellow. The acidity may also be tested by dipping a strip of blue litmus paper into the water. If it turns red, the water is acid. This method is not as sensitive as the previous one, which should be used in preference. If litmus paper is kept in stock, it should be kept in a bottle with a glass stopper, as exposure to the atmosphere will deteriorate the paper. If the water in the boilers has become corrosive and corrosion has set in, the water in the gauge glass will show red or even black. As soon as the color is beyond a dirty gray or straw color, it is advisable to introduce lime or soda to neutralize the acid.

To test for carbonate of lime, pour some of the water to be tested into an ordinary tumbler. Add a little ammonia and ammonium oxalate; then heat to the boiling point. If carbonate of lime is present, a precipitate will be formed.

When testing for sulphate of lime, pour some of the feedwater into a tumbler and add a few drops of hydrochloric acid. Add a small quantity of a solution of barium chloride and slowly heat the mixture. If a white precipitate is formed that will not redissolve when a little nitric acid is added, sulphate of lime is present.

When making a test for organic matter, add a few drops of pure sulphuric acid to the sample of water. To this add enough of a pink-colored solution of potassium permanganate to make the whole mixture a faint rose color. If the solution retains its color after standing a few hours, no organic substances are present.

Matter in mechanical suspension may be tested for by keeping a tumblerful of the feedwater in a quiet place for a day. If sediment is present in the tumbler, there is mechanically suspended matter in the water.

FEEDWATER HEATING

ECONOMY

54. It is important that the feedwater should be introduced into the boiler at as high a temperature as possible. The advantages of hot feedwater are: (1) The avoidance of the strains produced by the unequal expansion of the plates of the boiler by the introduction of cold feedwater; (2) the saving of fuel effected by the higher temperature of the feedwater. In order that there will be a direct saving of fuel, it is necessary that the heat used for heating the feedwater be taken from some source of waste; the principal ones being the waste furnace gases and the exhaust steam from the engine. When the feedwater is heated in an apparatus that utilizes the heat from the exhaust steam, or from live steam, it is called a **feedwater heater**, which may be installed either in the engine room or fireroom, as most convenient. When the feedwater is heated by waste gases from the furnaces, the heater is called an **economizer**, and it is placed in the flue between the boilers and the stack. Economizers have not been successfully applied to marine boilers as yet, except in connection with some of the pipe boilers, their great weight, the space occupied by them, and the cost of up-keep being greater disadvantages than the benefits derived from their use.

Feedwater heaters as used with condensing engines utilize some of the heat in the exhaust steam from the auxiliary engines, steam pumps, etc. only; the exhaust steam from the main engine going to the condenser. On modern ocean steamers, the auxiliary engines, steam pumps, etc. will furnish an ample supply of exhaust steam to heat the feedwater to near the boiling point. Feedwater heaters applied to non-condensing engines utilize some of the heat of the exhaust steam from the main engines, which would otherwise go to waste.

55. The economy of using hot feedwater may be shown by a simple calculation. Suppose that a boiler is required to

furnish steam at 145 pounds gauge pressure (160 pounds, absolute) and that the feedwater is introduced into the boiler from a condenser at a temperature of 100° F. The number of British thermal units required to change a pound of water at 100° F. into steam at 145 pounds gauge pressure is, from the Steam Table, about 1,125. Now, suppose that the feedwater was passed through a heater and its temperature raised to 210° F., at which temperature it enters the boiler, instead of at 100° F., as before. Then the number of British thermal units gained thereby are $210 - 100 = 110$, and the gain in per cent. is $\frac{110}{1,125} = .978 = 9.78$ per cent. Every increase of 10° F. in the temperature of feedwater effects a saving of approximately 1 per cent. of fuel.

CONSTRUCTION OF FEEDWATER HEATERS

56. A **Blake marine feedwater heater** is illustrated in Fig. 18. It is constructed on what is known as the jet system. This heater consists of two sections *a* and *b*, designated as the upper chamber and the receiver. The feedwater and the exhaust steam from the auxiliary engines, steam pumps, etc. are brought together in the upper chamber by means of the spray cone *c*, which is adjustable from the outside by the hand wheels *d*, *d'*. In the upper chamber, the heat of the steam is quickly absorbed by the feedwater. The water then falls to the receiver below, where it is allowed to accumulate only in sufficient quantity to operate the float *e*, which, by means of the levers, rods, etc. shown, controls the steam throttle valve *f* that supplies steam to the feed-pump. This valve is balanced and regulates the speed of the pump in a positive manner. The feedwater is pumped up from the hotwell tank into the upper chamber of the heater through the feedwater inlet *g*, the water spraying through the adjustable cone into the dome of the heater. After passing through the spray nozzle in the form of a thin sheet, the water is still further atomized by two perforated baffle plates. The exhaust steam from the auxiliary

engines, steam pumps, etc. enters the heater by the steam-exhaust inlet nozzle *h* through the automatic check-valve *i*.

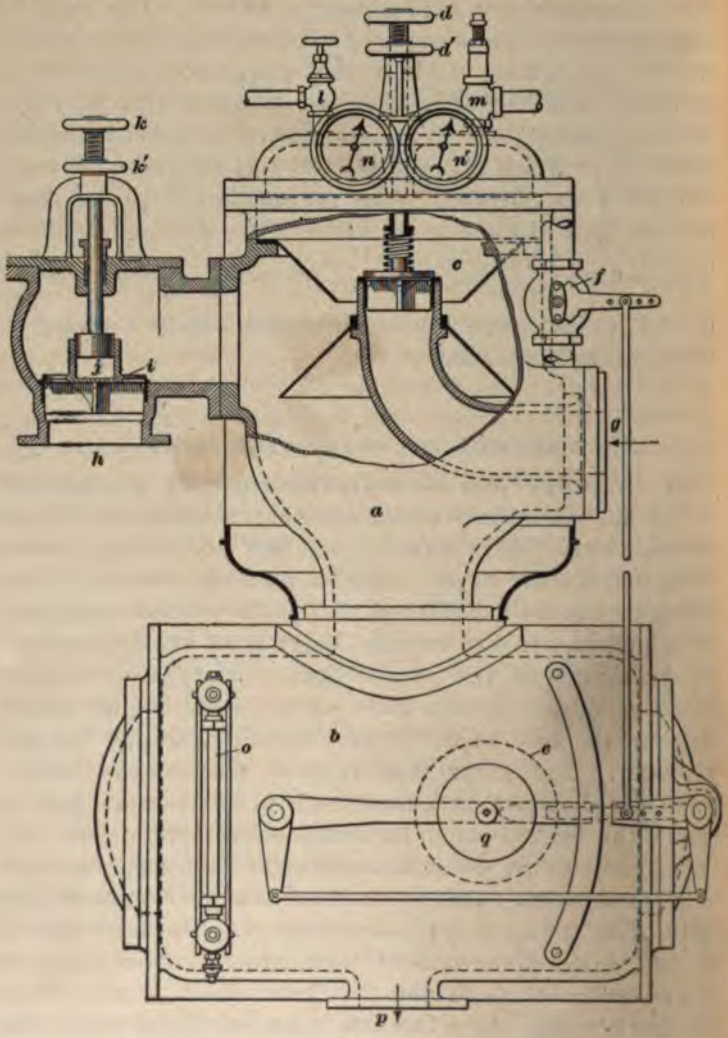


FIG. 18

This check-valve is provided with a dashpot *j* on its upper side. The amount of cushion for the dashpot is adjustable

om the outside by means of the hand wheels k, k'. This eater is also provided with the air valve l for allowing the r and uncondensed vapors to pass from the top of the heater the surface condenser. The safety valve m, also located 1 the top of the heater, can be set at any pressure desired—sually the pressure carried in the low-pressure receiver of e main engine. The usual practice is to have a branch pipe onnecting the heater with the receiver of the low-pressure ylinder of the engine, so that any surplus exhaust steam from e auxiliaries not condensed by the feedwater will pass to e engine—or vice versa. The heater is also provided with eam- and water-pressure gauges, shown at n and n'. The ass water gauge o shows clearly the level of the water in e receiver. The feedwater outlet to the pump is shown at p. e balanced steam throttle valve f is shown as being placed ongside of the heater, but it is very desirable to have this lve located as near the feed-pump as possible and concted to the ball-float lever by suitable rods, etc.

The advantage claimed for this heater is the presence of the ceiving chamber referred to. This receiver acts as a reserir, in which the water comes to rest, frees itself from vapors, d maintains a steady, even level, so that the ball float govning the speed of the feed-pump moves slowly and with the ist oscillating movement, thus avoiding the uneven motion which other forms of heaters are liable. The feed-pump is, erefore, prevented from entirely draining the heater. The ll float is counterbalanced by the counterweight q, and, ving to its being in a horizontal chamber, it has a radius of tion not possible in a vertical cylinder of reasonable size.

57. The heaters shown in connection with the doctor, ustrated in Fig. 3, are known as *open heaters* from the fact at the part of the heater which contains the feedwater is en to the atmosphere through the exhaust pipe. Such en heaters, used only with non-condensing engines, are jected to by some engineers on the ground that the oil ed in the cylinders is carried into the heaters by the exhaust eam, and that consequently at least some of it mixes with

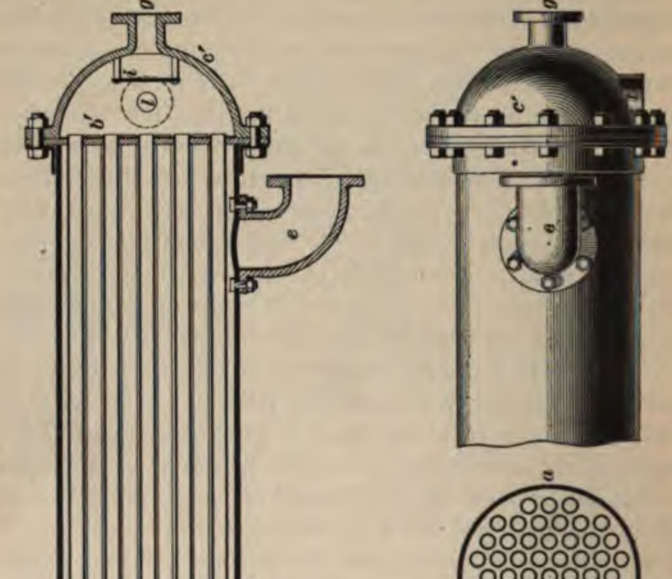


Fig. 19

feedwater and is carried into the boilers, where it deposits on the plates. It cannot be denied that this is true to some extent, but since river water is usually very muddy, the boilers must be frequently cleaned on account of the mud deposited; the oil carried in is then removed with the mud.

However, to overcome this objection, *closed feedwater heaters* have been designed. In these the feedwater passes through a closed system of pipes under pressure, is not exposed to the atmosphere at all during its passage from the suction pipe of the pump to the boilers, and does not come in contact with the exhaust steam. Hence, in the closed feedwater heater no oil can mix with the water.

58. One design of a closed feedwater heater for non-condensing engines, as used frequently on steamboats navigating the western rivers of the United States of America, is shown in Fig. 19. The heater consists of a cylindrical wrought-iron or steel shell *a*, to the ends of which angle-iron rings are riveted. The tube sheets *b*, *b'* and cast-iron heads *c*, *c'* are bolted to these rings. Tubes *d*, *d* are expanded into the tube sheets so as to form steam-tight and water-tight joints. The exhaust steam from the engine enters the nozzle *e* and leaves the heater through the nozzle *f*. It surrounds the tubes and heats the feedwater, which enters through *g* and leaves at *h*. The plate *i* serves to distribute the entering feedwater. Any condensed exhaust steam is carried off through the pipe *k* attached to the bottom of the heater; the water side of the heater can be emptied through a pipe attached at *l*. This style of heater is not adapted to a doctor of the description given, a pump that will simply force the water through the heater being all that is required. This pump handles cold water only, inasmuch as the water is heated after it leaves the delivery side of the pump.

The rules of the Board of Supervising Inspectors provide that the feedwater for a boiler used in connection with a non-condensing engine shall not be admitted at a lower temperature than 180° F. Hence, the necessity of employing a heater on river steamers will be apparent.

MARINE-BOILER FEEDING

(PART 2)

FEEDWATER

LOSS OF FEEDWATER

ORDINARY MEANS OF MAKING UP LOSS

1. When a surface condenser is used, a certain amount of water is evaporated into steam in the boiler, turned into water again in the condenser, taken from there to the boiler and reconverted into steam, used once more in the engine, again exhausted into the condenser, and so on. It will be seen that, if there were no loss through leakage and in other ways, the same water could be used over and over again, no further supply being needed. But, as a matter of course, there is always a certain amount of leakage going on, in the piping conveying the steam to the engines, in the engines themselves (at the glands, etc.), and in the feed-pumps and piping. Besides this, there is the loss due to blowing the whistle and to steam used in the various auxiliary engines that do not exhaust into a condenser. All these losses must be made good. This loss of water may be discovered by watching the water gauge of the boiler. Should the water gradually become less in the boiler, with the feed-pumps operating properly, it shows that there is a deficiency of water in the hotwell.

The most common way of making up for the loss consists in connecting the water end of the condenser with the steam

end of the condenser by means of a **U**-shaped pipe. This pipe has a valve or cock in it, which is opened whenever an additional amount of feedwater is required, and is allowed to remain open for a certain length of time. Some of the cooling water will flow into the steam side of the condenser and mingle with the condensed steam. The arrangement described is known as the **salt feed.**

When a ship is fitted with ballast tanks filled with fresh water, a small pipe provided with a stop-valve may connect the inside of the condenser with the tanks. In this case, when the stop-valve is opened, the pressure of the air forces the water in the tanks into the condenser, there being a partial vacuum within it.

EVAPORATORS

2. Purpose.—In modern sea-going vessels, especially if intended for long runs, the loss of feedwater is usually made up by means of an apparatus called an **evaporator,** which converts sea-water into fresh water by evaporating it and condensing the steam. Since the solid matter contained in sea-water cannot be vaporized at the same temperatures at which water can be transformed into steam, the condensed steam from sea-water is pure if condensed in a separate vessel, as the impurities are left in the vessel in which the sea-water is evaporated. While the water from an evaporator is pure or fresh, it is not well adapted for drinking purposes, except in an emergency, it having a peculiarly flat and bitter taste. It is eminently suitable, however, for boiler feeding.

3. Construction.—Three views of one form of the **Baird evaporator,** which is largely used in American sea-going steamships, are shown in Fig. 1. Like reference letters refer to like parts in the several views. The construction of this evaporator is as follows: The vertical cylindrical vessel *a*, closed at both ends, is provided with a coil *b*, made up of a number of boiler tubes bent into **U** shape and with their ends expanded into the tube-sheet *c*. There are two nests of these tubes, one above the other. The dished

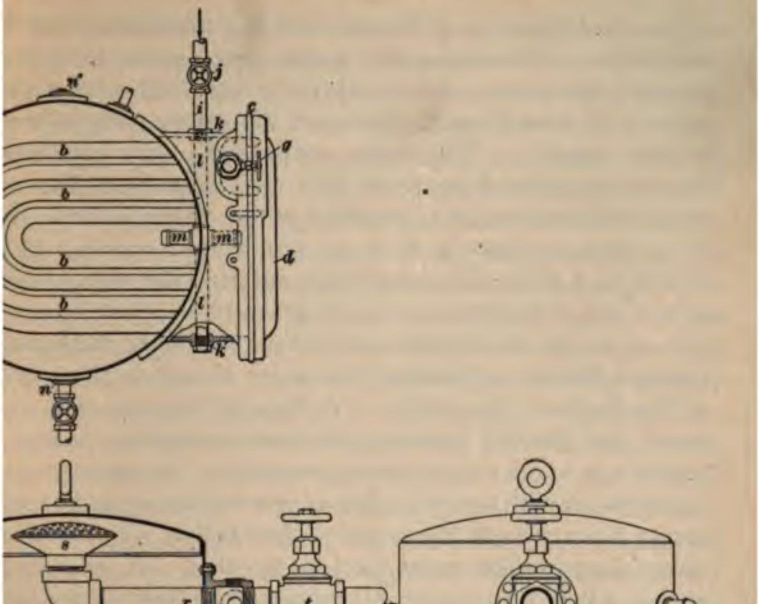


FIG. 1

cover, or bonnet, *d* is bolted over the tube-sheet, and has partitions in it, causing the steam that enters the space *e* through the steam pipe *f*, which is provided with a stop-valve *g*, to flow through the coil in the zigzag path indicated by the arrow *x*. The steam condensed in the coil leaves through the pipe *h* provided with a globe valve. The sea-water admission pipe *i*, provided with a globe valve *j*, may be connected either at *k* or *k'*, the unused opening being closed by a plug. The sea-water entering through *i* passes into a pipe *l* provided with a cross and two short pieces of pipe *m*, *m*, closed at their ends and perforated at the bottom, through which perforations the water passes to the bottom of the inside of the shell *a*. A blow-off pipe leading overboard, and serving to empty the shell *a*, may be attached to nozzle *n* or *n'*, as may be most convenient. A steam gauge *o* is connected with the coil *b*, and a combined steam and vacuum gauge *p* connects to the inside of the shell *a*. A glass water gauge *q* shows the water level in the shell. A large vapor pipe *r*, fitted with a rose *s*, connects the inside of the shell with the condenser; it is provided with a stop-valve *t*. The nests of tubes can be removed bodily for cleaning by unscrewing the nuts on the studs holding the tube-sheet *c* and bonnet *d* in place, and then withdrawing the nest from the shell.

4. The operation of the evaporator is as follows: The stop-valve *t* is opened, thus connecting the inside of the evaporator with the steam side of the condenser, and hence forming a vacuum inside the shell *a*. The salt-water admission valve *j* is now opened and sea-water is allowed to flow into the shell until the coil *b* is covered to a depth of several inches, as indicated by the glass water gauge *q*, when *j* is closed. Live steam is now admitted to the coil by opening the valve *g*, and the valve in the drain pipe *h*, which pipe leads to a device that permits the condensed steam to drain from the coil *b*, but prevents the passing of steam. The live steam heats and evaporates the sea-water surrounding the coil, the vapor passing to the condenser. Since the water in the evaporator is subjected to a very low pressure, by reason

of the shell being connected to the condenser, slightly less heat is required to evaporate the sea-water than would be the case otherwise. This, however, is merely an incidental advantage of connecting the evaporator to the condenser, the main object of the connection being the condensation of the vaporized sea-water without a separate apparatus. The rate of evaporation is regulated by the stop-valve *g*, partly closing it to reduce the pressure of the steam and hence its temperature, whereby the evaporation rate is reduced. The pressure in the coil is indicated by the steam gauge *o*. When the water gauge *q* shows that most of the sea-water has been evaporated, more is admitted by opening the stop-valve *j*. Owing to the salt and the other scale-making impurities contained in sea-water, the density of the water in the evaporator will quickly increase, and hence the evaporator must be blown off frequently. This is done by first closing the stop-valve *t*, which causes some of the sea-water to be evaporated into steam; the pressure soon begins to rise, as shown by the gauge *p*. The stop-valve of the blow-off pipe *n* is opened when a sufficient steam pressure has been reached, and the dense water in the evaporator is blown out. The evaporator is now ready for a fresh charge of sea-water, and the operation may be repeated.

5. During the process of evaporating the sea-water, a large proportion of the scale-making impurities in the water will be precipitated by the heat and will adhere tenaciously to the outside of the tubes in the form of scale. This scale, being a non-conductor of heat, decreases the efficiency of the evaporator as it accumulates on the tubes, and hence arises the necessity of occasionally removing it from the tubes. The salt will remain in solution in the water until the water becomes saturated with it, after which it will deposit in the form of solid salt on the bottom of the evaporator; and if the process were allowed to continue under these conditions it would eventually fill up the evaporator solid with salt, rendering the apparatus useless; hence, it is necessary to blow out the water before it becomes saturated.

It will be observed that there are two sources of fresh-water supply from this evaporator: (1) the water of condensation from the steam coil *b*; (2) the condensation of the vapor from the sea-water that passes into the condenser through the pipe *r*. The water from both sources eventually flows to the hotwell or feed-tank, whence it is pumped into the boilers as make-up feedwater.

6. An important part of the evaporator is a device called a **steam trap.** This device permits the water formed by the condensation of the steam in the coil *b*, Fig. 1, to drain

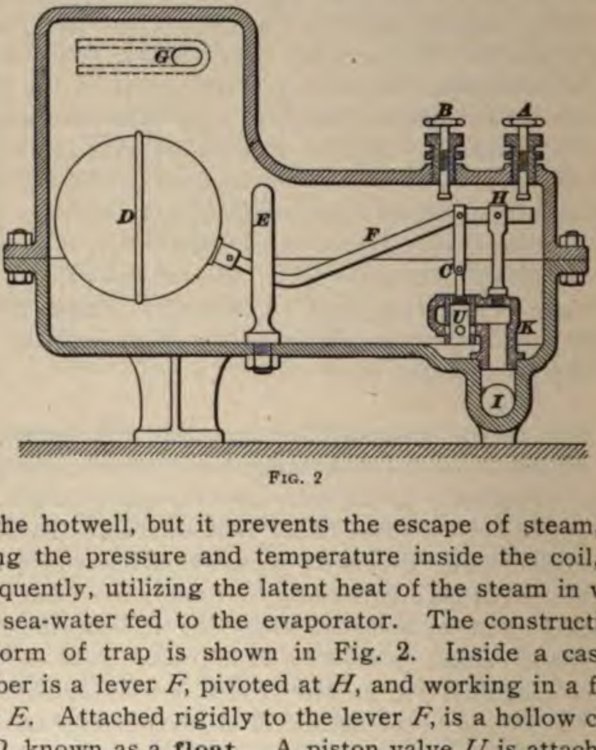


FIG. 2

into the hotwell, but it prevents the escape of steam, thus holding the pressure and temperature inside the coil, and, consequently, utilizing the latent heat of the steam in vaporizing sea-water fed to the evaporator. The construction of one form of trap is shown in Fig. 2. Inside a cast-iron chamber is a lever *F*, pivoted at *H*, and working in a forked guide *E*. Attached rigidly to the lever *F*, is a hollow copper ball *D*, known as a **float.** A piston valve *U* is attached by the link *C* to the lever *F*. The valve is provided with four ports, all of which, at a certain position of the valve, communicate with the passage in the valve chamber *K*, leading

to the drain pipe *I*, which communicates with the hotwell. The inlet pipe *G* is connected to the drain pipe of the evaporator. When steam is admitted to the evaporator, it flows through the coil into the drain pipe, thence into the steam trap, where further escape is prevented, since the outlet to the drain pipe *I* is shut off by the valve *U*, the ball *D* being in the position shown. The steam in the coil and the trap gradually condenses, the water gradually collects and rises in the trap and lifts the ball *D* until, by means of the connections shown, the valve *U* is opened, when the pressure of the steam will blow a certain quantity of the water through the four passages of the valve into the valve chamber, and thence into the drain pipe *I*. The ball *D* sinks as fast as the water leaves the trap, until the communication to the drain is shut off, when the water will again collect in the trap and the operation will be repeated. By means of the adjusting screw *A*, which limits the drop of the ball, the quantity of water discharged may be regulated. The screw *B*, forming a stop for lever *F*, is used for regulating the amount of opening of the valve.

Steam traps are made in a variety of forms; they are often used on the return pipes of the steam-heating systems of steam vessels and serve the same purpose as the one used in connection with the evaporator.

7. The **Quiggin evaporator** is illustrated in Fig. 3. Securely bolted to the inside of the vertical steel shell *a*, in the lower portion, are two annular, composition manifolds *b, b'*, of proper thickness to withstand the usual boiler pressure. Connected to these manifolds, in a vertical position, are the spiral-shaped tinned-copper heating coils *c, c*. The coils are interchangeable, and can be disconnected in a few minutes, when required. Owing to their inherent elasticity, it is difficult for them to leak, being unaffected by irregular expansion and contraction. If, by any mishap, the feed should be stopped, the coils will not be damaged by becoming exposed, and no harmful results to the evaporator can arise from inattention to the feed or from overheating the coils. Riveted

to the shell are two manhole frames, or a large door with a frame, so located as to afford ready access to the interior for cleaning purposes, or for disconnecting the coils, which can be done by means of an ordinary wrench. Fitted to the shell is a suitable plate of rolled zinc, of proper area, for taking up galvanic action. The shell being of the vertical type, and having an abundance of vapor space in the upper portion, priming is avoided.

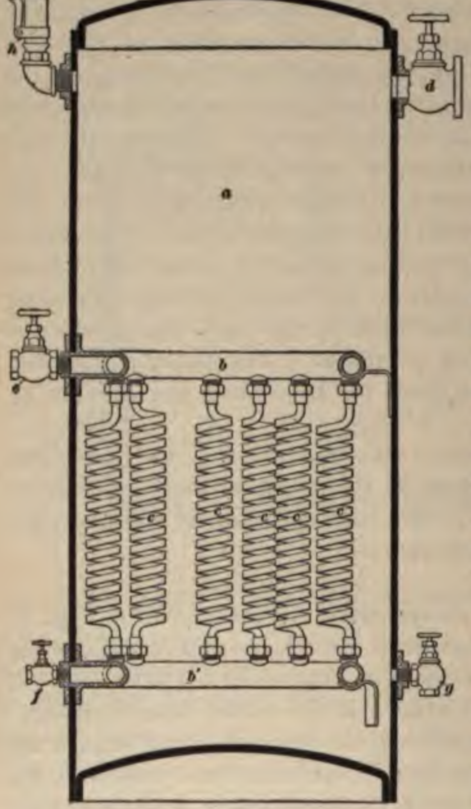


Fig. 3

The coils *c*, *c* in the lower part of the shell receive the steam from the main boilers, or from the intermediate-pressure receiver of the engine, through the upper manifold *b*. These coils are covered to a certain height by sea-water, which is fed into the evaporator by means of a small feed or donkey pump, set to supply the amount of water evaporated and to maintain a uniform feed-level of water in the evaporator. This pump takes its suction from the discharged circulating water, thereby getting the benefit of the heat that has been given up by the exhaust steam in condensing. The steam in passing through the coils gradually gives up its heat to the water surrounding them, and converts it into

vapor. By the time the steam reaches the bottom manifold b', it has given up all its heat above the temperature due to the pressure carried in the shell, and has been condensed to water, and in this form it flows to an automatic steam trap, thence to the feed-tank for the boilers. The vapor arising from the surface of the water comes in contact with the upper part of the coils and is thoroughly dried, and any spray or priming that might rise through too violent ebullition is thus prevented from passing over with the vapor through the valve d to the receiving tank, the condenser, or the low-pressure receiver.

The constant evaporation of the water, which leaves all the solids behind, necessarily causes an accumulation of scale on the heating surfaces, thus gradually reducing their efficiency. In this evaporator, the nature of the heating surface insures the cracking off of most of the scale as fast as formed, owing to the expansion and contraction of the spiral coils (the tubing of which has a crescent-shaped cross-section) that is always going on. The scale accumulates in the bottom of the shell and is readily taken out through the lower manhole. If any of the scale has hardened and still remains on the coil, it can be removed by blowing off the hot water and then turning on steam to the coils, thus causing a sudden expansion and breaking off of the scale; it can also be removed by striking the coils gently with a stick or hammer handle.

This evaporator will work practically automatically, and only requires attention to be given to the blowing off, and an occasional look at the feed-pump to see that it is not giving too much feed. It is nearly useless to blow out for the removal of sulphate of lime scale. This will form anyhow, and the more water blown out the more must be fed in. But, if the density is allowed to become too high, there will be a deposit of common salt. Blowing out will prevent this. Experience has shown that a density of $\frac{5}{32}$ may be safely carried. A higher density risks the deposit of salt, and a lower one means greater loss of heat.

When the evaporator is required for making up boiler feed only, it can be connected to the main exhaust pipe by means

of a spring regulating valve, which is supplied for the purpose by the manufacturer.

The evaporator is fitted with all the valves, gauges, and fittings necessary for its efficient operation. These comprise the vapor-outlet valve *d*, the steam-inlet valve *e*, the drain valve *f*, and the blow-off valve *g*. It is also provided with the safety valve *h*, a pressure gauge, and a glass water gauge (not shown in figure).

SALT MEASUREMENT AND REGULATION

MEASUREMENT

8. Saturation.—Ordinary sea-water contains on an average 1 pound of solid matter, about one-quarter of which is salt, in every 32 pounds of water. If sea-water is evaporated, the solid matter held in solution in the water remains; that is, if 32 pounds of sea-water is evaporated, 31 pounds of steam is formed and 1 pound of solid matter remains. Should but part of the 32 pounds be evaporated, say 16 pounds, there will remain 16 pounds of salt water containing 1 pound of solid matter. Again, if one-half of this is evaporated, the 1 pound of solid matter will still be contained in the remaining 8 pounds of salt water. Suppose that a vessel contains 32 pounds of sea-water, and that 16 pounds of water is evaporated, and there remains in the vessel 16 pounds of water containing 1 pound of solid matter. Another 32 pounds of sea-water is put into the vessel, and the same number of pounds are evaporated. It is evident that there is now 2 pounds of solid matter contained in the 16 pounds of water that is left in the vessel. This shows that the more sea-water is added and evaporated, the more solid matter will be contained in the water remaining in the vessel. This is exactly what takes place in a marine boiler using sea-water, and it is evident that, in order that the contained solid matter may not exceed a certain amount, a portion of the water must be occasionally drawn off from the boiler. This is done by means of either the bottom or the surface blow-off cock, and is termed **blowing off.**

The term **saturation** is used to denote the number of pounds of solid matter in every 32 pounds of water, and is usually expressed in the form of a fraction. Many engineers use the term **density** instead of saturation. For instance, $\frac{3}{32}$ saturation means that there is 3 pounds of solid matter in 32 pounds of water. Fresh water at sea level boils at 212° F., but if solid matter is added, the temperature of the boiling point will be raised. In Table I, the boiling points of sea-water at different degrees of saturation are given. As water will boil at a temperature varying with the pressure of the atmosphere, the boiling points given in the table are correct for but one pressure, namely, 30 inches of mercury.

TABLE I

BOILING POINTS OF SEA-WATER

Saturation	Boiling Point Degrees F.	Saturation	Boiling Point Degrees F.
$\frac{0}{32}$	212.0	$\frac{7}{32}$	220.3
$\frac{1}{32}$	213.2	$\frac{8}{32}$	221.5
$\frac{2}{32}$	214.4	$\frac{9}{32}$	222.7
$\frac{3}{32}$	215.5	$\frac{10}{32}$	223.8
$\frac{4}{32}$	216.7	$\frac{11}{32}$	225.0
$\frac{5}{32}$	217.9	$\frac{12}{32}$	226.1
$\frac{6}{32}$	219.1		

At $\frac{12}{32}$ saturation, the water becomes saturated; that is, it will not dissolve any more solid matter.

9. Finding Saturation by Thermometer.—It will be seen, by referring to Table I, that the boiling point rises about 1.2° F. for every pound of solid matter added. It has been determined that, for every tenth of an inch variation in the height of the barometer, the boiling point of the water varies .16° F. If the height of the barometer is less than 30 inches, the water will boil at a lower temperature than given in Table I. For instance, if the height of the barometer is 29 inches, water containing $\frac{2}{32}$ solid matter will boil

at $214.4 - 10 \times .16 = 212.8°$. Conversely, if the height of the barometer is 30.4 inches, the same water will boil at $214.4 + 4 \times .16 = 215.04°$ F.

Should it be desired to find the saturation of the water in the boiler, some water is drawn from it into an open vessel and is then heated to the boiling point. The temperature at which it boils is ascertained by means of a thermometer, and the boiling point with the barometer standing at 30 inches is calculated. The saturation is found by the approximate rule given below:

Rule.—*To find the degree of saturation by means of the thermometer, subtract 212° from the corrected boiling point of the water tested and divide the remainder by 1.2.*

Or,
$$S = \frac{A \pm B - 212}{1.2}$$

where S = saturation;

A = boiling point of water tested;

B = product of number of tenths of an inch variation in height of barometer and .16, this product to be subtracted when barometer is above 30 inches, and to be added when below 30 inches.

EXAMPLE.—A sample of water boils at 215.2° F., the height of the barometer being 30.5 inches. What is the saturation of the water?

SOLUTION.—Since the barometer reading is above 30 in., the value of B must be subtracted from that of A. Substituting values,

$$S = \frac{(215.2 - 5 \times .16) - 212}{1.2} = 2 \text{ lb. of solid matter, or } \tfrac{2}{32} \text{ saturation. Ans.}$$

10. Finding Saturation by Hydrometer.—It is evident that as the density of water increases, the more solid matter there is dissolved in it, and consequently, by measuring the density of the water, the amount of solid matter may be readily found. This is done by means of a **hydrometer**, shown in Fig. 4. This instrument is often called a **salinometer**; that is, a salt measurer, as it is used for measuring the quantity of salt contained in water. It consists of a glass tube, near the bottom of which are two bulbs. The lower and smaller bulb is loaded with mercury or shot, so as to cause the instrument to remain in a vertical

when placed in the water. The upper bulb is filled and its volume is such that the whole instrument is an an equal volume of water. Most salinometers ated to read off the density when the water has a are of 200° F.; some of these instruments, however, e scales, one each for 190° F., 200° F., and 210° F.

A salinometer is graduated by trial, placing it first water having the temperature at, say, 200° F. The which the instrument sinks in the water is marked on tube. It is then placed in water having the same ure and containing $\frac{1}{32}$ part of salt. The depth to instrument now sinks is the sea-water mark, and marked on the tube. This operation is with water containing $\frac{2}{32}$, $\frac{3}{32}$, and so on part of salt, always taking care that the ure of the water is exactly 200° F. The the tube are transferred to a paper scale, pasted to the inside of the tube in exactly position as the marks on the tube. The between the marks are usually divided es, quarters, and eighths. If the sali- is to give a correct reading at 190° F. F., the process of graduating must, of e carried out at the desired temperature ut. Knowing the process of graduation, neter can be improvised out of a piece of thin tall and slender bottle, etc.

FIG. 4

f the hydrometer is placed in a vessel containing er drawn from the boiler and having a temperature hat at which the instrument was graduated, it will depth corresponding to the density of the water, saturation may be read off on the scale. For if the hydrometer sinks to the $2\frac{7}{8}$ graduation on the means that there is $2\frac{7}{8}$ pounds of solid matter in pounds of water.

the temperature of the water under test vary from erature at which the hydrometer was graduated, the

indication of the hydrometer will not be correct. This is due to the difference in density of water at different temperatures. Allowance may be made for this error in the following manner: The indication of the hydrometer will vary $\frac{1}{80}$ part of 1 pound of solid matter for every degree the temperature of the water varies from the temperature at which the hydrometer scale was marked.

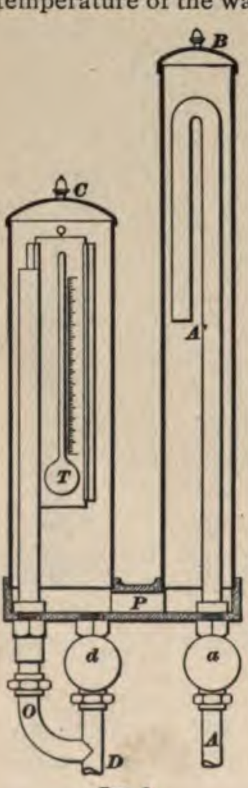


FIG. 5

Thus, if the temperature is 205° F., a hydrometer graduated at 200° F. will show $\frac{5}{80}$ part of 1 pound of solid matter less than there really is in the water. For instance, the temperature of the water being 205° F., and the hydrometer indicating $1\frac{15}{16}$ pounds of salt, the actual amount of salt will be $\frac{5}{80}$ of 1 pound more, and as $\frac{5}{80} = \frac{1}{16}$, $1\frac{15}{16} + \frac{1}{16} = 2$ pounds will be the actual saturation. Also, if the temperature of the water is less, the indication of the hydrometer will be too high. For example, the temperature of the water being 180° F., and the hydrometer indicating $2\frac{1}{4}$ pounds, the indication will be $\frac{20}{80}$ pound too high. Subtracting $\frac{20}{80} = \frac{1}{4}$ from $2\frac{1}{4}$, 2 pounds is found to be the true amount of solid matter in every 32 pounds of the water drawn from the boiler.

13. For convenience, a **salinometer pot**, shown in Fig. 5, is commonly used. It is attached to the boiler, or, should there be several boilers, it is put up in the engine room and connected by branch pipes with each boiler. The pot consists of two cylindrical brass vessels, one of larger diameter than the other, communicating with each other by a passage P in the base of the instrument. A removable cover C is fitted to the larger vessel. The pipe A connects

th the water space of the boiler. On opening the stop-lve a, water flows into the smaller vessel and through the ssage P into the larger one. The water is prevented from erflowing by the overflow pipe O, which empties below : drain valve d into the drain pipe D. A thermometer T licates the temperature of the water. If the temperature the water is too low, the stop-valve a is opened until the sired temperature is reached; if too high, the water is owed to cool. To prevent the water admitted to the pot ing upwards and scalding the attendant, the admission e A' is turned downwards, as shown. Any vapor formed y escape through the perforated top B. When the water the larger vessel is at the proper temperature, the density iscertained by means of the hydrometer, and then the pot lrained by opening the drain valve d.

REGULATION

4. It is evident that, to keep the water in the boiler at ertain degree of saturation, the solid matter carried with feedwater into the boiler in a stated time must be ioved from the boiler in the same length of time; that is, number of pounds of feedwater multiplied by the amount olid matter in 1 pound, must equal the number of pounds vater blown off multiplied by the amount of solid matter l pound.

n rules I and II, the quantity of water may be taken in nds, tons, gallons, cubic feet, etc., but it is absolutely essary to use the same denomination throughout the cal-ition.

.et A = quantity of water blown off;
B = saturation of A;
C = quantity of feedwater;
D = saturation of C;
E = quantity of water evaporated, corresponding to C.

tule I.—*To find the quantity of water to be blown off in a ed time, divide the product of the quantity of feedwater*

admitted in that time and its saturation by the saturation of the water to be blown off.

Or, $$A = \frac{CD}{B}$$

EXAMPLE 1.— 2,000 pounds of feedwater enters a boiler every hour at a saturation of $\frac{1}{32}$; how much water must be blown off every hour to keep the saturation at $\frac{2}{32}$?

SOLUTION.—Applying rule I,

$$A = \frac{2{,}000 \times \frac{1}{32}}{\frac{2}{32}} = 1{,}000 \text{ lb.} \quad \text{Ans.}$$

EXAMPLE 2.—How much water must be blown off every hour to keep the saturation at $\frac{3}{32}$, when 48,000 gallons of sea-water, at a saturation of $\frac{1}{32}$, is fed to the boiler in 24 hours?

SOLUTION.—The water fed per hour $= \frac{48000}{24} = 2{,}000$ gal. Applying rule I,

$$A = \frac{2{,}000 \times \frac{1}{32}}{\frac{3}{32}} = 666\tfrac{2}{3} \text{ gal.} \quad \text{Ans.}$$

Rule II.—*To find the quantity of water evaporated into steam for a given quantity of water blown off, divide the saturation of the water blown off by the saturation of the feedwater, subtract 1 from this quotient and multiply the remainder by the quantity of water blown off.*

Or, $$E = A\left(\frac{B}{D} - 1\right)$$

EXAMPLE 3.—The saturation of the feedwater being $\frac{1}{32}$, how much water is evaporated into steam for 1 pound of water blown off at $\frac{2}{32}$ saturation, the saturation to be kept constant?

SOLUTION.—Applying rule II,

$$E = 1 \times \left(\frac{\frac{2}{32}}{\frac{1}{32}} - 1\right) = 1 \text{ lb.} \quad \text{Ans.}$$

Rule III.—*To find the quantity of water blown off for a given quantity evaporated, divide the saturation of the water blown off by the saturation of the feedwater, and subtract 1 from the quotient. Divide the quantity of water evaporated by the remainder.*

Or, $$A = \frac{E}{\frac{B}{D} - 1}$$

EXAMPLE 4.—The saturation of the boiler being kept at $\frac{5}{32}$ and the feedwater being at $\frac{1}{32}$, how much water must be blown off for every pound of water evaporated?

SOLUTION.—Applying rule III,

$$A = \frac{1}{\frac{\frac{5}{32}}{\frac{1}{32}} - 1} = \tfrac{1}{4} \text{ lb. Ans.}$$

The total feed obviously must be the sum of the water evaporated and the water blown off, in order to keep a steady boiler water level. Thus, if the evaporation is 6,000 pounds per hour, the feedwater at $\frac{1}{32}$, and the boiler worked at $\frac{4}{32}$ saturation, the amount blown off per hour, by rule III, is $\frac{6{,}000}{\frac{\frac{4}{32}}{\frac{1}{32}} - 1}$ $= 2{,}000$ pounds, and the total feed is $6{,}000 + 2{,}000 = 8{,}000$ pounds per hour.

15. It was formerly the practice not to let the saturation of the water in the boilers exceed $\frac{2}{32}$. This limit to saturation has, however, been gradually raised to $\frac{5}{32}$, as less scale-forming matter is carried into the boiler at that density than at the lower one. Furthermore, with the water in the boiler at $\frac{5}{32}$, there is sufficient difference between the specific gravity of the oil or grease carried in by the feedwater, even when combined with some of the carbonate of lime, and that of the boiler water to insure the grease and oil floating on top, whence it can largely be removed by a frequent use of the surface blow-off. Since much less scale-forming material is carried into the boiler at a high density, there is proportionately less scale formation.

The reason that less scale-forming material is carried into the boiler when the water is at a high density is that less water requires to be blown off and replaced. Besides this, there is the additional advantage in a high saturation of less waste of heat.

16. A simple calculation will show that less solid matter is carried in at a high saturation than at a low one. Suppose that a boiler contains 100,000 pounds of sea-water at its steaming level, and evaporates 50,000 pounds of water per

hour. Let sea-water at $\frac{1}{32}$ be used for boiler feeding, let the saturation be kept at $\frac{2}{32}$, and let the steaming period be 6 days of 24 hours each.

At the beginning, the boiler contains $\frac{100000}{32} = 3{,}125$ pounds of solid matter. To bring the saturation to $\frac{2}{32}$, 100,000 pounds of water must have been evaporated and replaced, thus bringing in 3,125 pounds more of solid matter. The time required to bring the saturation to $\frac{2}{32}$ is $\frac{100000}{50000} = 2$ hours, at the end of which period the boiler contains $3{,}125 + 3{,}125 = 6{,}250$ pounds of solid matter. By rule III, Art. **14,** the quantity blown off per hour is $\dfrac{50{,}000}{\frac{\frac{2}{32}}{\frac{1}{32}} - 1} = 50{,}000$ pounds, giving a total feed per hour of $50{,}000 + 50{,}000 = 100{,}000$ pounds, which brings in $\frac{100000}{32} = 3{,}125$ pounds of solid matter. In 6 days there is $6 \times 24 = 144$ hours, of which 2 hours was consumed in bringing the saturation to $\frac{2}{32}$, leaving $144 - 2 = 142$ hours, during which $142 \times 3{,}125 = 443{,}750$ pounds of solid matter is carried in. The total solid matter carried in during 144 hours is $443{,}750 + 6{,}250 = 450{,}000$ pounds, under the assumed conditions.

Now consider the same boiler with the saturation kept at $\frac{5}{32}$. To bring the saturation to $\frac{5}{32}$, the water has to be changed four times; that is, $100{,}000 \times 4 = 400{,}000$ pounds must be fed in and evaporated, leaving behind $\frac{400000}{32} = 12{,}500$ pounds of solid matter. With the first filling, 3,125 pounds was carried in. The time required to evaporate 400,000 pounds of water is $\frac{400000}{50000} = 8$ hours, at the end of which period the boiler contains $3{,}125 + 12{,}500 = 15{,}625$ pounds of solid matter. By rule III, Art. **14,** the quantity blown off per hour is $\dfrac{50{,}000}{\frac{\frac{5}{32}}{\frac{1}{32}} - 1} = 12{,}500$ pounds, giving a total feed per hour of $50{,}000 + 12{,}500 = 62{,}500$ pounds, which brings in $\frac{62500}{32} = 1{,}953.125$ pounds of solid matter. This amount is carried in during $144 - 8 = 136$ hours, during which time $136 \times 1{,}953.125 = 265{,}625$ pounds of solid matter is carried in, making a total for 144 hours of

265,625 + 15,625 = 281,250 pounds, against 450,000 pounds carried in when the saturation is kept at $\frac{2}{32}$.

17. By drawing hot water from the boiler and making up the deficiency with colder water, a certain amount of heat is lost. The amount is equal to the difference in temperature between that of the water in the boiler and that of the feedwater. For instance, if the temperature of the water in the boiler is 320°, and that of the feedwater 180°, then 320° − 180° = 140° will be the amount of heat lost. This represents the number of British thermal units lost for each pound of water blown off and replaced with a pound of cooler water. Knowing this, the difference in the loss of heat due to blowing off at different densities can be found. To keep the density at $\frac{2}{32}$, by rule III, Art. **14**, for each pound of water evaporated there must be blown off $\dfrac{1}{\dfrac{\frac{2}{32}}{\frac{1}{32}} - 1}$ = 1 pound of water, the density of the feedwater being $\frac{1}{32}$. If the feedwater temperature is 160°, and the temperature in the boiler 300°, the loss of heat per pound of water evaporated is 300 − 160 = 140 British thermal units. All conditions remaining the same as before except that the saturation in the boiler is kept at $\frac{5}{32}$, the water blown off for each pound evaporated is $\dfrac{1}{\dfrac{\frac{5}{32}}{\frac{1}{32}} - 1} = \frac{1}{4}$ pound, and the loss of heat per pound of water evaporated is $140 \times \frac{1}{4} = 35$ British thermal units. This shows that the heat lost by blowing off is reduced by carrying a high density.

18. The percentage of loss of heat due to blowing off is found as follows:

Rule.—*To find the percentage of loss of heat due to blowing off, divide the number of British thermal units lost in blowing off 1 pound of water by the sum of the total heat (above the temperature of the feedwater) imparted to the amount of water converted into steam for 1 pound of water blown off, and the number of British thermal units lost in blowing off 1 pound of water.*

Or,

$$L = \frac{B}{A + B}$$

where L = percentage of heat lost;

A = total heat (reckoned above temperature of feed) imparted to amount of water converted into steam for 1 pound of water blown off;

B = number of British thermal units lost in blowing off 1 pound of water.

EXAMPLE.—The saturation of the water in a certain boiler is to be kept at $\frac{3}{32}$; the temperature of the feedwater being 160° F., its saturation $\frac{1}{32}$, and the steam pressure 90 pounds, absolute, what will be the percentage of loss due to blowing off?

SOLUTION.—The sensible heat of steam at 90 lb. pressure is 320.094° F.; 320.094° − 160° = 160.094°, represents the difference of temperature, and the number of B. T. U. lost in each pound of water blown off. In order to find the total heat mentioned in the rule, the amount of water converted into steam for 1 pound of water blown off has first to be found from rule II, Art. **14**. This amount is found to be 2 lb. The total heat above 32° F. of a pound of steam at 90 lb. pressure is 1,179.569 B. T. U., and above 160° F. it is 1,179.569 − (160 − 32) = 1,051.569 B. T. U. Consequently, for 2 lb. the total heat is 1,051.569 × 2 = 2,103.138 B. T. U. For each pound of water blown off, 1 lb. will have to be fed into the boiler, in addition to the amount evaporated, and the amount of heat carried into the boiler by this pound of feedwater must be added to the total heat. Now, substituting the values,

$$L = \frac{160.094}{2{,}103.138 + 160.094} = .0707 = 7.07 \text{ per cent., nearly. Ans.}$$

EXAMPLES FOR PRACTICE

1. The height of the barometer being 30.4 inches, find the boiling point of water having a saturation of $\frac{5}{32}$. Ans. 218.54° F.

2. A sample of water boils at 216.7° F., the height of the barometer being 30.3 inches; find the saturation. Ans. $\frac{2.58}{32}$

3. A salinometer graduated for 200° F. immersed in water having a temperature of 208° F. indicates 3 pounds of salt; correct the reading. Ans. 3.1 lb.

4. How much water at $\frac{3}{32}$ saturation will have to be blown off to keep the saturation constant, the saturation of the feedwater being $\frac{1}{32}$, and 5,000 pounds entering the boiler every hour? Ans. 1,666.67 lb.

5. How much water at $\frac{1}{32}$ is evaporated for every 120 gallons of water blown off at $\frac{3}{32}$? Ans. 240 gal.

6. Find the percentage of heat lost in blowing off water at $\frac{4}{32}$, the steam pressure being 100 pounds, absolute, the temperature of the feedwater 190° F., and its saturation $\frac{1}{32}$. Take the total heat of 1 pound of steam at 100 pounds pressure, absolute and above 32 F°, as 1181.866 British thermal units, and its sensible heat as 327.625° F.

Ans. 4.29 per cent., nearly.

HEAT TRANSFER TO WATER

NATURAL CIRCULATION

19. The transfer of heat from the furnace to the water in the boiler is accomplished by radiation, conduction, and convection. It is estimated that when the fire is burning brightly, about one-half of the heat received from the furnace by the boiler is radiated. The transfer of heat through the water is due to convection, since liquids are poor conductors of heat. The particles of water next to the shell or plate become heated, and immediately rise into the main body of water, giving place to fresh particles of cold water. This setting up of a current by the action of heat is called **circulation.** The rapidity with which heat will be absorbed by convection depends on the effectiveness of the water circulation in the boiler, and on the extent and conductivity of the heating surfaces. The transfer of heat through the shell and furnace plates takes place by conduction. It has been proved, experimentally, that the quality or thickness of the material has little influence, thick iron tubes working practically as well as thin brass ones. Very thick plates are, however, liable to be injured by burning when exposed to the direct action of the fire.

20. Water circulation is essential to the efficient operation of a boiler. It has just been stated that the rapidity of the transfer of heat by convection depends on the rapidity of circulation. Besides this, the circulation is useful in preventing, in some degree, the deposit of sediment that accumulates from the feedwater. Again, a rapid circulation keeps the parts of the boiler at a uniform temperature.

21. Fig. 6 shows the circulation of the water in an externally fired cylindrical boiler. The heated currents rise from the hottest part of the shell directly over the furnace, and carry the bubbles of steam to the surface. The cooler water rushes in to take their place over the furnace, and thus the circulation is maintained. As shown in the figure, there are two currents, one carrying the cold water from rear to

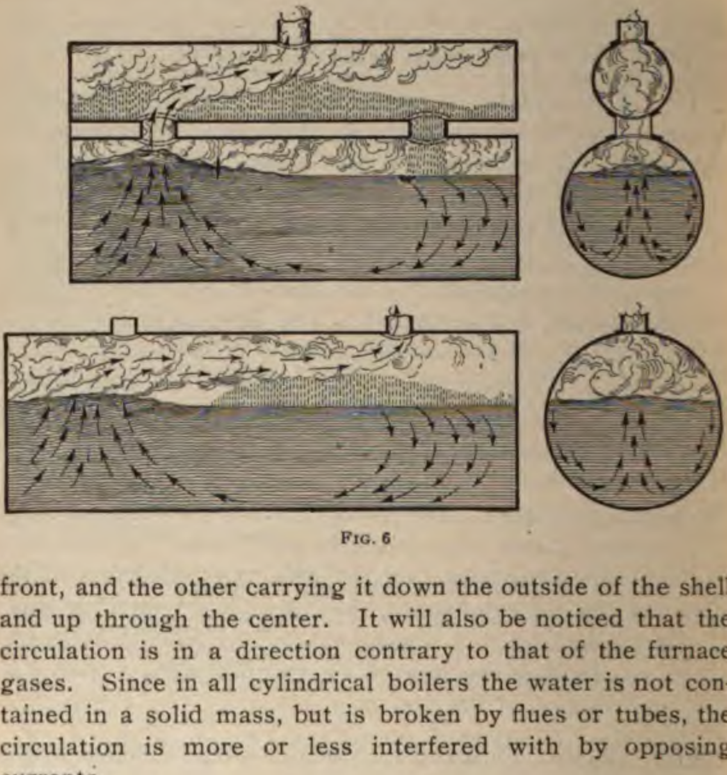


FIG. 6

front, and the other carrying it down the outside of the shell and up through the center. It will also be noticed that the circulation is in a direction contrary to that of the furnace gases. Since in all cylindrical boilers the water is not contained in a solid mass, but is broken by flues or tubes, the circulation is more or less interfered with by opposing currents.

22. Fig. 7 illustrates the circulation of the water in a Scotch boiler. The water directly over the top of the furnaces is heated first; it rises between the tubes and alongside each nest of tubes, cooler water coming down between the nests and alongside the shell. The circulation in the

lower half of the boiler is very feeble; some of the water passes upwards alongside the furnaces, where the downward currents meet it and almost neutralize its motion. It will

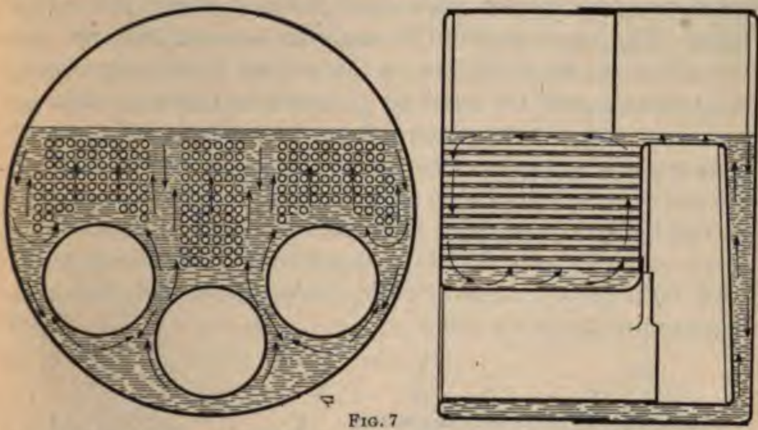


FIG. 7

thus be seen that the circulation is greatly interfered with by opposing currents, and as a consequence the lower half of the boiler is considerably cooler than the upper half; the difference in temperature between different parts of the sheets sets up severe strains in the material, thus tending to shorten the life of the boiler. The circulation is more rapid and effective if the water is constrained to follow a particular path. To accomplish this object, thin sheet-iron plates are sometimes fitted in Scotch boilers in such a position that the upward and downward currents cannot interfere with each other. The arrangement of the plates is shown in Fig. 8. The thin iron plates *P*, *P* enclose each separate nest

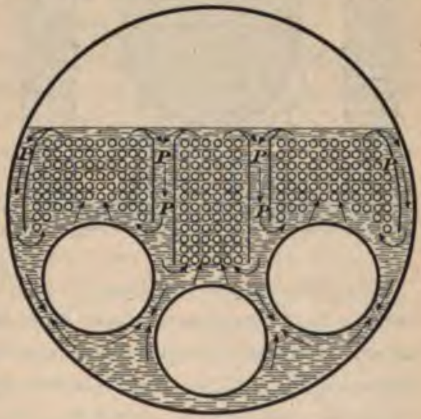


FIG. 8

of tubes, extending very nearly the whole length, and downwards to the lowest tubes of each nest. The upper ends of the plates enclosing each nest are inclined toward each other, and are carried up a short distance above the water level. The water is lifted through the opening between the two plates by the ebullition on the surface of the water, and, after parting with the particles of steam suspended within it, flows down the upper inclined surfaces of the plates and augments the downward current. These plates will accelerate the circulation in a direction at right angles to the axis of the boiler, but do not influence any current in an axial direction. The arrows show the direction of the currents. This arrangement helps the circulation somewhat, but still leaves much to be desired.

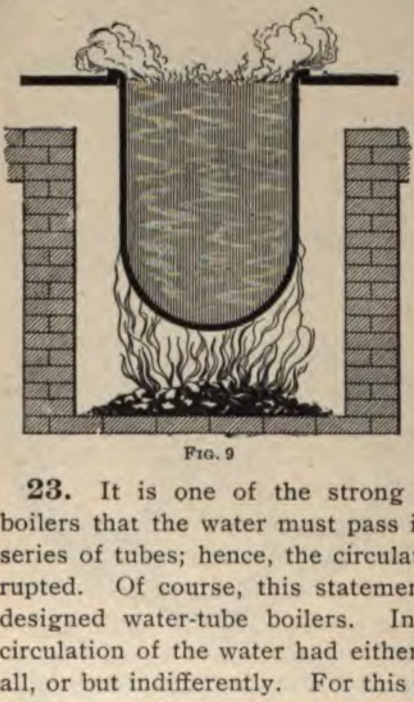


FIG. 9

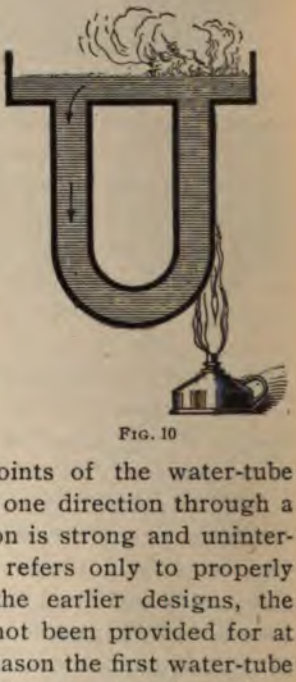


FIG. 10

23. It is one of the strong points of the water-tube boilers that the water must pass in one direction through a series of tubes; hence, the circulation is strong and uninterrupted. Of course, this statement refers only to properly designed water-tube boilers. In the earlier designs, the circulation of the water had either not been provided for at all, or but indifferently. For this reason the first water-tube boilers placed in a steamship were failures; but since then, the importance of providing for a rapid circulation has been

recognized and the boilers designed accordingly, so that at the present day the statement made holds good for nearly all water-tube boilers used in steam vessels. The difference between the cylindrical and water-tube boilers in this respect may be illustrated as follows: The cylindrical boiler with its contained mass of water may be compared to an ordinary kettle in the process of boiling (see Fig. 9). The water rises rapidly around the outer edges and flows downwards in the center. If, however, the fire is quickened, the upward and downward currents interfere with each other, and the kettle boils over. The water-tube boiler should be identical in principle with a **U** tube hanging from a vessel filled with water, and with the heat applied to one leg (see Fig. 10). The circulation is set up immediately, and proceeds quietly, no matter how fierce the fire may be.

FORCED CIRCULATION

24. Of late years, much attention has been paid to the improvement of the circulation of the water in the fire-tube marine boilers, and today a great many of them are fitted with some apparatus for improving the circulation. This may be done by a small pump connected to the bottom of the boiler, drawing the cold water from the lower half and discharging it through a perforated pipe near the water level downwards between the nests of tubes.

25. The **Craig heating and circulating apparatus** is shown in Figs. 11 and 12. In general design and principle of operation, it greatly resembles an injector. In Fig. 11, the feedwater enters through the pipe L, passing through the check-valve C into the nozzle B; thence through A into the feed check-valve M, and through the stop-valve R into the boiler. The water passing through B at a high velocity induces a current of water to pass through the induction pipe N connected to the bottom of the boiler, the water passing through the check-valve E into D, thence through the annular opening between the nozzles F and G into F_1, whence it passes into A, mingling with the feedwater. To heat the water, a

jet of live steam is admitted by the valve K and nozzle G. The steam pipe P, bolted to the valve K, leads to the donkey boiler. When it is desired to equalize the temperature and circulate the water while getting up steam in a boiler fitted with the apparatus, steam from the donkey boiler is admitted by the valve K. The steam flowing through G into F at a high velocity acts the same as an injector, inducing a flow of water through N, E, D, F, A, M, and R into the boiler, and

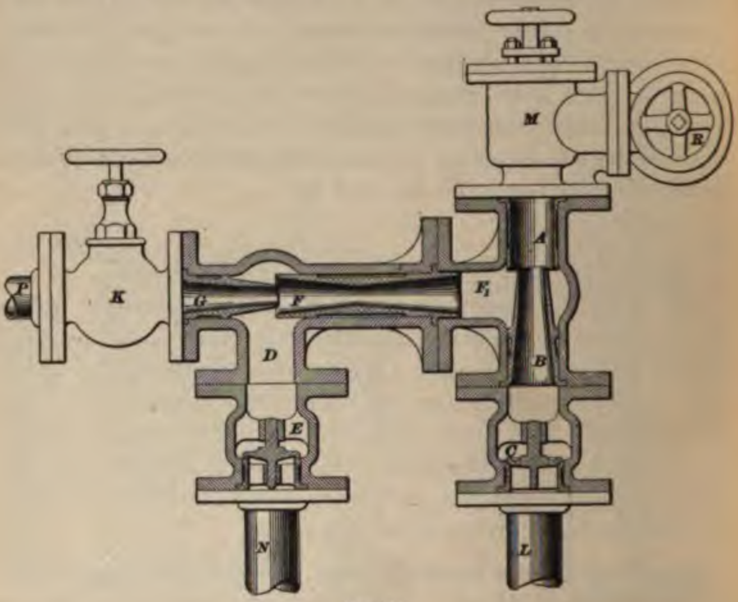


FIG. 11

delivering the water at a high temperature. The check-valve C prevents any of the water from entering the feed-pipe L. When the boiler is under steam, the apparatus is used to circulate and heat the feedwater, the circulation being induced by the feedwater flowing through B with a high velocity and inducing a current of water to pass up through N, E, D, F, and F_1 into A, and thence into the boiler. The feedwater is heated by mingling with the hot water coming from the bottom of the boiler.

26. In Fig. 12, the apparatus is shown applied to a Scotch oiler. At N_1, the internal induction pipe is shown. This a perforated pipe running in the direction of the length of ie boiler, and is connected to the external induction pipe N, rovided with a stop-valve n. The feedpipe L is provided ith a stop-valve l. At M the feed check-valve is shown,

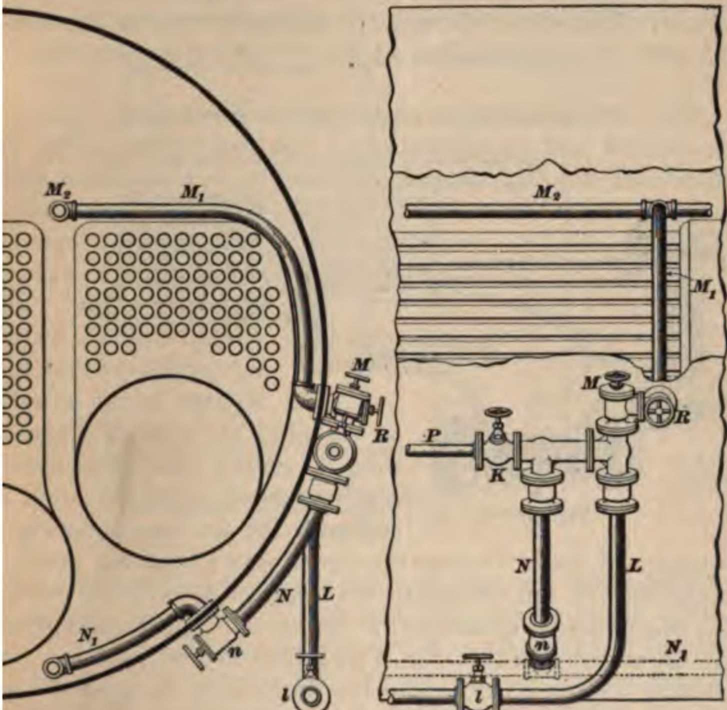


FIG. 12

hence the water passes through the stop-valve R into the iternal feedpipe M_1, thence into the perforated distributing ipe M_2, discharging the water downwards between the nests f tubes. P is the steam pipe leading to the donkey boiler. 'his pipe is provided with a stop-valve K. To circulate ie water while running, the valve n is opened, when the eedwater itself will maintain the circulation. To heat and

circulate the water before fires are lighted, the valve *l* is closed, the valves *M* and *n* are opened, and steam from the donkey boiler is admitted to the apparatus, the steam, in the manner previously explained, heating and circulating the water, thus bringing all the parts of the boiler to a uniform temperature, and thereby greatly reducing the local strains in the material of the boiler due to starting a fire in the furnaces. The apparatus may be connected by suitable piping to serve for the main feed as well as for the donkey feed.

27. Bloomsburg's equilibrium circulator is shown in Fig. 13, and its application to a marine boiler in Fig. 14. Referring to Fig. 13, the feedwater enters at *a* in a solid body, and in flowing through the annular opening *b* assumes a tubular shape. The whole device being immersed in water, the friction of the annular jet issuing at a high velocity from *b* causes the surrounding water to move in the direction of and with the jet, thus inducing a current of water to flow through *f* into *d* and out at *e*.

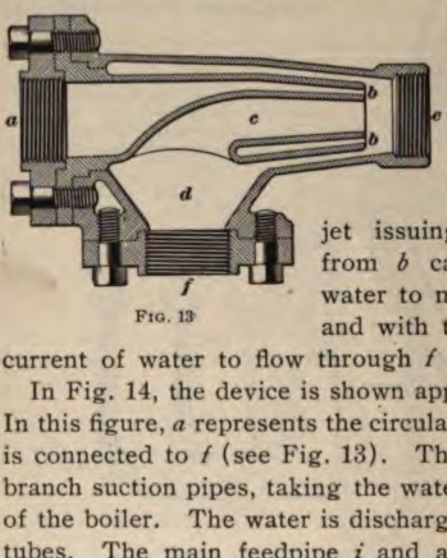


FIG. 13

In Fig. 14, the device is shown applied to a Scotch boiler. In this figure, *a* represents the circulator. The suction pipe *g* is connected to *f* (see Fig. 13). This suction pipe has two branch suction pipes, taking the water from the coolest part of the boiler. The water is discharged through *h* above the tubes. The main feedpipe *i* and auxiliary feedpipe *j* are both connected to the circulator (at *a* in Fig. 13). From the foregoing it is seen that as long as the feed-pumps are working this device will automatically improve the circulation.

In order to improve the circulation while getting up steam, and also in order to heat the water in the boiler, a jet similar to that shown in Fig. 13 is sometimes placed at the junction

of the main and branch suction pipes, the jet pointing upwards into the main suction pipe. By means of a suitable pipe connection and valve, live steam from the donkey boiler or one of the main boilers can be turned into the jet, thus inducing a current of heated water to flow upwards. By means of this supplementary device, circulation can be kept up and improved when the feed-pumps are not working, or it can be used in conjunction with the circulator if desired.

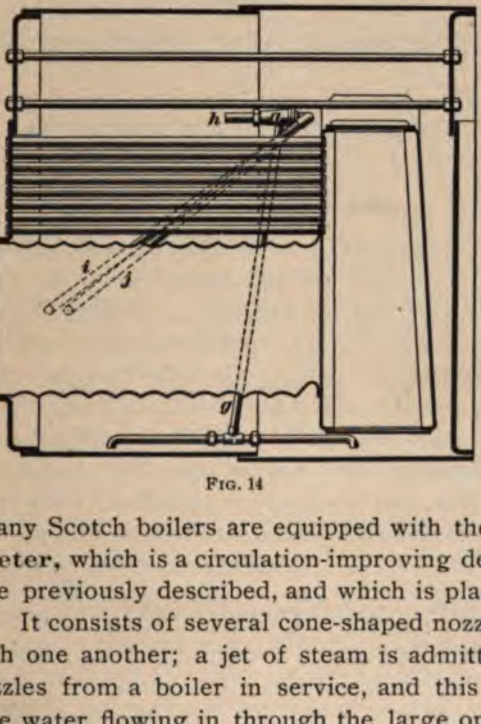


FIG. 14

28. Many Scotch boilers are equipped with the so-called **hydrokineter,** which is a circulation-improving device similar to those previously described, and which is placed inside the boiler. It consists of several cone-shaped nozzles placed in line with one another; a jet of steam is admitted axially to the nozzles from a boiler in service, and this induces a current, the water flowing in through the large open end of the nozzles and out of the small part. The hydrokineter is used chiefly in getting up steam in a cold boiler, giving a forced circulation.

MARINE-BOILER MANAGEMENT

MANAGEMENT WHEN STEAMING

GETTING READY FOR SEA

GENERAL INSTRUCTIONS

1. Examination of Boilers and Fittings.—The Rules and Regulations of the United States Steamboat Inspection Service state as follows: "It shall be the duty of an engineer, when he assumes charge of the boilers and machinery of a steamer, to forthwith thoroughly examine the same, and if he finds any part thereof in bad condition, caused by neglect or inattention on the part of his predecessor, he shall immediately report the facts to the local inspectors of the district, who shall thereupon investigate the matter; and if the former engineer has been culpably derelict of duty, they shall suspend or revoke his license."

2. In making the required examination of the boilers, on assuming charge, the engineer should inspect them both internally and externally. Before entering a boiler that has just been opened, it should be tested for foul air by holding a lighted lamp or candle inside of it and noting the effect on the flame. If the flame burns brightly, it will be safe to enter the boiler; but if it burns dimly and finally goes out, the boiler should not be entered until it has been thoroughly ventilated. If the donkey boiler, or any other boiler on board, has steam in it at the time the examination is being made and there is a steam connection between it and the

boiler undergoing inspection, the stop-valve in that connection should be tightly closed and secured so that it cannot be opened by mistake, thus preventing the possibility of scalding the man making the inspection.

The principal defects to look for are *bulges*, *cracks*, *blisters*, and *thin* and *burnt places* in the plates composing the heating surfaces. The first defect named is revealed by an ocular inspection; the others can be discovered easiest by tapping the plates with a light hammer and listening for any difference in sound. The seams, rivets, staybolts, and tubes should be examined for leaks. The stayrods should be calipered to ascertain if they have been reduced in thickness, to any great extent, by corrosion; their ends should be examined to see if they are properly secured. The fusible plugs should also receive attention. The thickness of the scale on the tubes and on the plates composing the heating surfaces should be noted, and whether there are any large flakes of rust peeling off them; also, note if there is much mud or other sediment in the boilers. Examine the crown bars. Look at all openings to pipes, gauges, safety valves, etc. to see that they are clear. Look the dry pipe over and see that the perforations or slits are clear. Examine the zinc protectors and see that they are properly placed and connected, and that the baskets are in good condition. Look very carefully for oil and grease on the heating surfaces, especially on the crown sheets and the top sheets of the combustion chambers; if any is found, be sure to have it removed before the boiler is closed up. Before coming out of the boiler, look for lamps, oil cans, bunches of waste, tools, and other foreign matter liable to be left there by the workmen.

After completing the examination in the steam and water space, the heat space should be entered and inspected with the same degree of care and thoroughness that was given to the water space and steam space. Such defects as bulges, cracks, blisters, thin and burnt places in the plates, and leaks in seams, rivets, staybolts, and tubes can best be detected from the fire side of the plates. Leaks will generally be

revealed by having more or less salt or rust around them, or by bare spots where the soot has been blown away by jets of steam or water squirting through the leaks. The crown sheets and the back sheets of the combustion chambers should receive special attention, as it is on these sheets that the flames impinge, and no part of a boiler is exposed to more intense heat and greater strains. While inside the heat spaces, examine the bridge walls, grate-bar beams and their lugs, and also the dead plates. The exteriors of the boilers and front connections should now receive attention. Examine the boiler coverings and note if there are any leaks in, or rusty spots on, the shells. Look over all the pipes that are connected to the boilers, especially the feedpipes, back to the source of the feedwater supply, inclusive of the feed-pump and all cocks and valves in the feedpipes, as well as those in all the other pipes. Try the dampers, safety valves, gauge-cocks, cocks in glass water gauge, and water-column connections. Examine the salinometer pots and connecting pipes; also, all drain pipes, and try their cocks.

If any defects are found during the examination, they should be remedied at once; and if the boilers require cleaning and scaling, there should be no delay in having these operations performed. If any rusty places are found on the shell of the boilers, the rust should be scraped off and the bare spots painted with red-lead paint. Pack all valve stems that need it; in fact, put everything in good order so as to be in readiness to raise steam whenever required to do so.

3. Fireroom Force.—On an ocean steamer, the fireroom force consists of the water tenders, firemen, and coal passers. There is usually one fireman and one coal passer to every 100 or 125 square feet of grate surface in the boilers. In the United States Navy, the men on duty in the fireroom are under the immediate charge of the water tender, who also regulates the feedwater supply, the steam pressure, etc. In the merchant service, these duties are usually performed by an engineer; and on small vessels, the engineer of the watch attends to them. The fireroom force is usually divided

into three divisions, called **watches**, each division taking its turn and being on duty 4 hours and off duty 8 hours. This rotation goes on continuously day and night while the vessel is under way. In vessels plying regularly between ports only a short distance apart, there are often but two watches, of 6 hours each; and in vessels making very short runs, there is usually but one watch.

4. Coaling Ship.—Before taking in coal, the bunkers should be examined; and if any of the braces have been removed to facilitate taking out coal during the last run they should be replaced; all rubbish should be removed from the bunkers and the doors closed. On vessels in which coaling ports are not provided, the scuttle plates on deck should be taken off and the chutes rigged. The required number of shovels, slice bars, coal mauls, and bunker lamps for stowing the bunkers should be sent on deck; also, if considered necessary, a weighing scale for weighing the coal. A competent man should be detailed to run the winch, and other men to *tally*, that is, count, the tubs or baskets of coal as they come on board. Men should also be detailed to stow the coal and trim the bunkers. After the coaling has commenced, a sufficient number of tubs or baskets of coal should be weighed to get their average nėt weight, and after the coaling has been completed the number of tubs or baskets of coal put into the bunkers multiplied by their average net weight will give the total amount of coal received. When the bunkers are nearly full, the stowing of the coal should be carefully attended to, so that there will be no empty spaces left in the bunkers. After the bunkers are filled, the chutes are unrigged and stowed away, the scuttle plates put on, the shovels and other tools are collected and sent below. The winch engine is wiped off and drained, and the cover put on, if one is provided.

GETTING UP STEAM

5. General Preparations.—Clean off manhole and handhole cover-plates and their seats and renew gaskets wherever necessary. Replace the grate bars. See that the

blow-off cocks, the drain cocks, and valves in pumping-out pipes are closed. Ease off the main and auxiliary stop-valves and seat them gently. Open cocks in connections to water columns, glass water gauges, and steam gauges. Close cocks in the pipes to the salinometer pots. Open feed stop-valves. Remove smokestack hood. Slack off the smokestack guys. Open the damper. Examine the valves of the auxiliary feed-pump, and overhaul the pump if it requires it. Try the ash hoist.

6. Closing Boilers.—When the engineer of the watch receives orders to get up steam, he immediately summons to the fireroom the division of men whose turn it is to go on watch and assigns to each fireman the furnaces he is to take care of and details a coal passer to supply him with coal.

On the arrival of the men in the fireroom, they proceed at once to close up the boilers by putting on the manhole and handhole cover-plates. A thin coating of black lead (graphite) and tallow mixed together should be spread over the gaskets if they are of sheet rubber or asbestos. If corrugated copper gaskets are used, the black lead and tallow may be dispensed with.

7. Filling Boilers.—If the vessel is lying in fresh and pure water, it is only necessary to open the bottom blow cocks and let the water flow in to its proper level from overboard, the safety valves having previously been opened to permit the escape of air as the water flows in. Should the boilers be so located in the vessel that the water from overboard will not rise to the required level, the amount lacking must be pumped into them. If the vessel is lying in impure, very muddy, or salt water, the boilers should be filled from some other source. If the vessel is lying at a wharf, the boilers may be filled by a hose attached to the water pipes on shore. If the boilers are not fitted with nozzles for the purpose of attaching the hose, the upper manhole plate of each boiler should be left off and the hose inserted through the manholes. If the vessel is lying in the stream, it will be necessary to get the water supply from a water boat.

8. Starting Fires.—While the water is running into the boilers, the furnaces may be charged. The rear halves of the grates are covered with a thin layer of coal and the front halves with split cord wood. Some of the wood should be broken into kindling, which is placed under and amongst the front ends of the sticks of cord wood just inside the furnace doors; then a bunch of oily waste, or shavings, if any are on hand, is put amongst the kindling and ignited. When the wood is burning freely, coal is thrown on top of the wood, a little at a time. By the time the wood is all consumed, there will be thin beds of live coals all over the grates. The fires are then built up gradually by throwing thin layers of coal on top of the burning coal until the fuel bed reaches the required thickness.

9. Raising Steam.—When getting up steam, the fires should not be forced, but should be allowed to burn up gradually, thus giving the boiler an opportunity to expand more uniformly under the influence of the increasing heat. By forcing the fires, the plates or tubes that are nearest the fires are subjected to extreme expansion, while those parts that are remote from the fire are still cold; under such conditions, the seams and rivets, and also the tube ends, are liable to be severely strained, and possibly permanently injured.

It is not desirable to raise steam in an internally fired fire-tube marine boiler in less than from 3 to 5 hours, while from 7 to 9 hours, and even more, would be better. Externally fired fire-tube boilers are not subjected to the strains due to expansion to such an excessive degree as the internally fired type, and water-tube boilers are still less affected by unequal expansion and contraction; therefore steam may be raised in such boilers without serious injury in less time than in internally fired boilers.

Assuming that the pressure at which the boilers are to operate has been reached, before connecting them with the engine, all the cocks and valves should be tried under pressure. The safety valves should be raised for a moment and their action noted; the water columns should be blown

through and the gauge-cocks tested; the feed stop-valves should be opened and the feed-apparatus tried; and it should be particularly noted whether the check-valves seat properly. The blow-off cocks should also be tried and their condition noted. Everything being found in good condition, the boilers will be ready for service.

10. Smokestack Guys.—While in port, the smokestack guys may either be cast off at their lower ends and the slack coiled down on the fireroom-hatch gratings near the pipe, or they may be kept in their places and hauled taut. If the latter method is practiced, the guys should be slacked off before getting up steam, and in either case, after the smokestack has expanded to its full height, the guys should be hauled taut and the ends firmly secured, so that they will support the smokestack when the ship rolls and pitches at sea.

LEAVING PORT, AT SEA, AND COMING TO

GETTING UNDER WAY

11. When the signal to get ready to start the engines is received in the engine room, the boiler attendant should be notified at once; he should then immediately open the damper, if closed, close the furnace door and connection doors, if any of them are open, and if the boilers are fitted with lever safety valves, and they are open, he should close them. The stop-valves in the main feedpipes should now be opened and those on the auxiliary feedpipes closed. When the steam pressure in the various boilers has risen to within, say, 10 pounds of the usual working pressure, the boilers are connected by opening their stop-valves. Before connecting, great care must be exercised to have the steam pressures in the several boilers practically equal. The stop-valves, in fact any valve that is subjected to great pressure, should be opened very slowly to prevent too sudden a change in the temperature and expansion of the piping through which the steam flows, and to prevent *water*

hammer. The latter is caused by large bodies of condensed steam being driven violently forwards by the out-rushing steam, due to opening a valve too quickly. Water hammer is liable to prove disastrous to the piping, the heavy blow due to the momentum of the body of condensed steam moving with high velocity being likely to cause a leaking of the joints, if not a bursting of the pipe. To prevent the accumulation of water, the steam-pipe drains should be kept open until the pipe is thoroughly warmed up; that is, until nothing but steam issues from the drains. In large vessels, with many boilers and long steam mains, it requires considerable time to thoroughly warm these pipes by a slow circulation of steam, and not until then should the boiler stop-valves be opened wide.

It is the practice of some engineers to open the main stop-valve entirely before warming up the steam piping; others warm up the piping as far as the main stop-valve and fully connect the boilers before, very slowly and by degrees, opening the main stop-valve. In the latter case, this valve should be very slightly moved from its seat before the boiler stop-valves are opened, in order that expansion may not jam it so hard that it cannot be opened. Neither practice possesses any great advantage over the other.

As soon as the steam pressure at the throttle has risen to the desired point, the engineer will commence to warm up the engines. Care must be exercised in the boiler room not to let the steam pressure run up high enough to lift the safety valves; this involves a careful watching of the steam gauges and subsequent regulation of the fires, checking a too rapid steam generation by putting the ash-pit dampers in place, by the main damper, and, as a last resort, by opening the furnace doors and finally the front-connection doors.

WORKING THE FIRES

12. The fuel bed should be kept at an even thickness. As a general rule, this should be about 8 or 10 inches, though this thickness may have to be varied to suit the different kinds and grades of coal used and the intensity of the draft.

The surface of the fire should be kept level and only enough coal should be put on at one time to fairly but thinly cover the entire surface of the glowing coals. No lumps of coal larger than a man's fist should be put on a fire if the best results are expected. Care should be taken to prevent holes being burned in the fuel bed. If there are any thin spots in the fire or if the surface is uneven it should be leveled off before coaling.

13. After a fire has been burning a certain length of time, it will require cleaning, as all varieties of coal contain more or less clinker-making material. The time and method of cleaning a fire depend principally on the nature of the fuel used and the rapidity with which it is consumed.

14. When a fire is permitted, by carelessness or otherwise, to get so low that it will no longer make steam, the best way to build it up is to draw all the live coals to the front end of the furnace, haul out all ashes, clinkers, and dead coal, and cover the grate bars back of the live coals with fresh coal. If soft coal is used, the fire will soon work its way back through the fresh coal and ignite it; but if anthracite is used there will be more difficulty experienced in building up a low fire. If an anthracite fire should get very low, the best method is to haul it and start a new fire with wood, or with live coals from one of the other fires. It is useless to throw wood on top of a nearly burned-out anthracite fire, as that course will make matters worse.

15. The best course to pursue to hold the steam pressure in check when the engines are stopped temporarily is to close the damper and ash-pit doors. If the pressure still continues to rise, the bleeder may be used to work off the surplus steam. The **bleeder** is a pipe of fairly large size connecting the main steam pipe directly with the condenser, permitting the surplus steam to be condensed, and also preventing the noise that the safety valves would make when discharging. This pipe is generally fitted in naval vessels, where condensers with independent air pumps and circulating pumps are the rule. If the stoppage of the engines is

prolonged beyond a few minutes some of the fires may be pushed back from the front, uncovering the grates for a short distance, thus checking the formation of steam. The following should be avoided, if possible, as they are all objectionable or detrimental to the boilers: blowing off with the safety valves; pumping in cold water and blowing with the blow-off cocks; opening the furnace doors and connection doors. When the engines are started again, the damper should be opened, the bleeder shut off, and if the fires have been pushed back, they should be spread out again as quickly as possible.

16. It is necessary to systematically remove the ashes from the ash-pits and to dispose of them, as well as of the refuse drawn from the furnaces in cleaning fires, in a suitable manner. In vessels where the fireroom force is divided into watches, it is the usual custom to clear the ash-pits and firerooms of all ashes and clinkers near the end of each watch. When the ashes have to be hoisted from the fireroom, they are usually wetted down to prevent excessive dust. In wetting down ashes, they should not be deluged with water from a hose or buckets, on account of the cloud of fine ashes that arises when so treated, which afterwards settles on everything within its reach. The proper way to wet down ashes is by means of a spray nozzle on the ash hose. If the vessel is provided with an ash ejector, the ashes may be ejected overboard whenever convenient to do so, and much trouble and annoyance may thereby be obviated.

17. In ships navigating the high seas, ashes are disposed of by throwing them overboard; in harbor and river navigation, regulations generally prohibit the throwing overboard of ashes and other refuse. Ashes must then be collected in suitable iron cans and removed from the ship at suitable intervals by ash and garbage collectors, who dispose of them in such a manner as the municipal or other regulations demand. As this service must be paid for, it is the practice in ships navigating the high seas to throw overboard all ashes and other refuse just before entering the harbor limits, thereby reducing the expense of ash and garbage removal.

PRIMING AND FOAMING

18. Priming.—Priming in a steam boiler is analogous to the boiling over of the water in a teakettle. On the application of intense heat, the water in contact with the bottom of the kettle or the heating surfaces of a steam boiler will be rapidly converted into steam, which will rise to the surface with considerable force, carrying the water with it until it overflows the kettle, or, in the case of a boiler, carrying the water into the steam pipe and thence into the engine cylinders, where it is liable to do considerable damage if it is not checked in time. There are several causes for priming, of which the most common are the following: (1) insufficient boiler power; (2) defective design of the boiler; (3) water level carried too high; (4) irregular firing; (5) sudden opening of stop-valves.

When the boiler power is insufficient, the boilers must be forced in order to furnish enough steam for the engine, and, consequently, the steam bubbles will rise through the water with such speed that they will carry particles of water with them by friction and cohesion. The best remedy for this cause of priming is to install larger boilers or more of them; the next best course to pursue is to put in a steam separator, which, obviously, will only prevent the entrained water from reaching the engines; it will not stop the priming.

Defective design in boilers generally consists of too small steam space, or of an imperfect arrangement of tubes, which may be spaced so close together in an effort to obtain greater heating surface as to seriously interfere with the circulation of the water. In some cases, a small steam space can be increased by the addition of a steam drum; or the top row of tubes may be taken out to advantage, which will admit of carrying a lower water level and thus increase the steam space. Defective water circulation is difficult to detect and to remedy; it may be due to too close spacing of the tubes, a marked improvement having occasionally been effected by the removal of one or two vertical rows of tubes. Thin sheet-iron plates are sometimes fitted in Scotch boilers in

such a position that the upward and downward currents of water cannot interfere with each other. Mechanical circulators are now largely applied to marine boilers with beneficial results.

The remedy for too high a water level is obvious—carry the water at a lower level. With irregular firing, especially when the draft is strong, the rate of evaporation will be so high at times that the steam bubbles will rise at such speed as to carry the water with them, just as in the case of insufficient boiler power.

The sudden opening of a stop-valve or the throttle valve causes a momentary local lowering of the pressure near the steam outlet of the boiler; consequently, some of the water in the other parts of the boiler will, by the greater pressure, be thrown toward the outlet and mix with the steam that is rushing from the boiler.

19. Priming manifests itself by a peculiar cracking sound in the cylinders of the engines, due to the water being thrown violently against the heads. In cases of very violent priming, the water will rise several inches in the glass gauge, thus showing a false water level. When priming takes place, one method of checking it temporarily is as follows: Close the damper and ash-pit doors, thereby checking the fires until the water has quieted down. The throttle or the main stop-valve should also be partly closed to check the onrush of water from the boilers and also to increase the pressure on the surface of the water, which tends to keep it from rising. After the priming is checked, observe if the water level drops in the glass water gauge, as it probably will; if it does, more feedwater will be required. To prevent damage to the engines, the drains in the steam pipe and cylinders should be opened. Regular and even firing tends to prevent priming by maintaining a steady pressure. Priming may also be produced by the main steam pipe being attached to the boiler too close to the water surface, or the absence of a dry pipe may offer an inducement for water to enter the steam pipe.

20. Foaming.—The changing of a body of water, varying in depth, at the normal steaming level of a boiler into foam is called **foaming**. This phenomenon shows itself in the water gauge glass by a violent and abnormal agitation and the absence of a well-defined water-line. It is extremely difficult to draw a sharp line of demarkation between foaming and priming, as the causes that produce priming may also produce foaming, and the same remedies will stop it in most cases. Foaming, may, however be caused by an excess of soda introduced in the boiler, by dirty or greasy water, and also by running from salt to fresh water, or vice versa, provided that the make-up feedwater is taken from overboard. If the engines are suddenly stopped while the boilers are foaming, the safety valves or bleeder should be immediately opened so as to keep the water foaming, otherwise, when the water level falls, some portions of heating surfaces may be uncovered and burnt. When foaming is caused by dirty or greasy water, much of the dirt and grease may be got rid of by blowing it out through the surface blow.

GENERAL BOILER MANAGEMENT AT SEA

21. Feedwater Regulation.—Each marine boiler is supplied with feed check-valves and feed stop-valves for the main feed and for the auxiliary. As a general rule, the check-valves are of the adjustable-lift type; in that case, the amount of water entering each boiler is regulated by varying the lift of the check-valve. If the check-valves are non-adjustable, the feed stop-valve is used for regulating the water supply. In practically all sea-going vessels, feed-pumps are run so as to keep the water level in the hotwell constant, and consequently they deliver practically the correct quantity of water required for the boilers. This fact permits the water tender to so adjust, by trial, the feed-valves that the water level in all the boilers will remain practically constant. When the water level in all the boilers is gradually dropping while that in the hotwell remains constant, it shows the need of additional feedwater, which is taken from

the salt feed, fresh-water tanks, or evaporator, according to circumstances. A gradual rising of the water level in all the boilers indicates that an undue quantity of water is entering the hotwell, most likely through the salt feed being partly open, or through burst condenser tubes.

22. Saturation Regulation.—With surface condensing engines in perfect order, very little, if any, salt water need ever enter the boilers, if the ship is fitted with an evaporator or ample fresh-water tanks. In this case, there is little need of testing the saturation of the water in the boilers more than once or perhaps twice a day, as at least once every four hours the engineer on watch should test, by tasting, the water in the hotwell to discover if any sea-water reaches the steam side of the condenser. When this is found to be the case, and it cannot be remedied before reaching port, the saturation will have to be tested quite frequently to note the rate at which it increases and thus enable the engineer in charge to determine whether blowing off will have to be resorted to before making port. Generally speaking, it is not necessary to blow off until the saturation reaches $\frac{4}{32}$ or $\frac{5}{32}$. In blowing off, only a relatively small amount of water can be blown off at a time, as the water level must not fall below the highest point in contact with the gases of combustion; the water level must then be restored by pumping, and the process of blowing off and pumping is repeated until the saturation has been sufficiently lowered. The greatest vigilance must be exercised by the person having charge of the blowing off to see that the blow-off cocks are fully closed after each blowing down.

23. Trimming Ventilators.—Ventilators are provided on board ship for the double purpose of conveying air to the furnaces and ventilating the fireroom, and it is important that they should be kept trimmed to the wind, while running, to supply the necessary amount of air. The tops or cowls of ventilators are made so that they may be turned toward the direction from which the wind is blowing, and they can usually be operated from the fireroom. If the vessel changes

its course or the wind shifts, the cowles are turned so that the wind will blow into them.

24. Finding Water Level.—The only safe and reliable method of testing the height of the water in a boiler is by means of the gauge-cocks. The glass gauge is merely a convenience, and it is not intended that it should be depended on exclusively, as it is very liable to become stopped up with dirt or scale and thereby show a false water level in the boiler. The glass gauges should be blown through frequently and their use should be supplemented by the frequent opening of the gauge-cocks.

25. Care of Safety Valves.—To test the freedom of working of safety valves, they should be lifted by means of the hand gear at least once a day. Should any safety valve be found to be stuck fast, the boiler to which it is attached should at once be cut out of service and the repairs made, unless the boiler is fitted with a second safety valve of sufficient capacity and in good order, in which case repairs can usually be deferred until the vessel reaches port.

Safety valves will occasionally leak, and thus waste fresh water, through a chip of scale or some other foreign substance lodging on the valve seat. This can usually be dislodged by lifting the valve, when the steam will blow the obstruction away. A leaky safety valve should be repaired by scraping and grinding at the earliest opportunity, as it not only wastes steam, which means fresh water, but is also a great annoyance owing to the noise it causes.

26. Care of Coal Bunkers.—A careful watch should be kept on the temperature of the coal bunkers. Coals that contain sulphur, especially when wet, are very liable to ignite from spontaneous combustion. If the temperature of a bunker should rise suddenly, it is an indication that spontaneous combustion is in progress, and the bunker should be carefully examined. If the coal is found to be on fire, it may be extinguished by the fire apparatus, or by steam, if pipes are fitted for that purpose, as is the case on board modern seagoing vessels. Coal bunkers should be ventilated

whenever the weather conditions on deck will admit of this being done without getting water into the bunkers. Wet coal should never be stowed in a ship's bunkers. Bituminous and semibituminous coals are more susceptible to spontaneous combustion than is anthracite.

27. Low Water in Boilers.—There is no more important duty connected with the management of boilers than to guard against letting the water get dangerously low in any of the boilers. The principal causes of low water in boilers are: insufficient feedwater, priming, leaky boilers, improper regulation of the check-valves, irregular firing, and sudden stopping of the engines.

Insufficient feedwater is the result of shortness in the supply of make-up feedwater, and the remedy is to increase that supply, whether it be from overboard, from tanks, or from an evaporator.

Low water caused by priming is the result of the water being carried over into the engines, the remedy being to check the priming and increase the feed.

Leaks in boilers may cause low water if they are numerous or extensive. When low water results from this cause, put on all the feed that is necessary to keep the water in sight in the glass gauge, and stop the leaks at the earliest opportunity.

If the check-valves are not properly adjusted, obviously the remedy is to regulate them so that each boiler will get the amount of water that it requires.

Irregular firing will cause a boiler to require more feedwater at one time than another; this will necessitate frequent adjustment of the check-valves, which if not carefully attended to will result in that boiler getting too much feedwater at one time and not enough at another time.

When the main engines are running at full speed, the water level in the boilers is usually higher than when the engines are at rest; and when the engines are suddenly stopped, the water-line almost instantly falls to its true level, which may be below the gauge cocks. When this occurs, the safety

valve or bleeder, or both if necessary, should be opened so as to keep the water lifting until the water level is pumped up above the danger point.

When the water level in a boiler drops below the gauge cocks or out of sight in the gauge glass, its exact location is unknown. Some portion of the heating surface may be uncovered and become very hot. If such be the case, it will be unsafe to put on the feed, as that act might result in the explosion of the boiler; the fires should be deadened as quickly as possible by throwing wet ashes or fresh coal on them, the boiler cut out of service, and then allowed to cool. The fires may be hauled after they have been thoroughly deadened. After the boiler is cool enough to be entered, a thorough examination should be made of it, noticing particularly the high parts of the heating surface. If the boiler is not seriously damaged, the fires may be started again, steam raised, and the boiler put into service. But if any serious injury, such as the cracking or burning of any of the plates, has been sustained, the boiler must remain out of service until it is repaired.

28. Importance of Carrying a Uniform Pressure. In order to obtain the best results from the engines and boilers for the amount of coal burned, it is necessary that the engines be kept running at a steady speed. This requires a uniform pressure, which can be maintained only by regular firing and a steady, continuous feed. A certain proportion of the fires should be kept in the best possible condition to make steam while the other fires are being worked. Two fires in the same boiler should not be worked at the same time. A routine of firing, or a system of working the fires by which each fire will be worked at regular intervals in its turn, should be inaugurated.

29. Sweeping Tubes While Under Way.—When the coal used is very fine and dry and of a variety that does not coke, large quantities of soot and coal dust collect in the tubes of fire-tube boilers, which reduce the draft area, thereby affecting the draft unfavorably. Besides, soot is a

non-conductor of heat, and a coating of it on the heating surfaces of a boiler will retard the passage of the heat into the water and permit a large part of the heat to escape up the smokestack and thus be lost. When this occurs, it is necessary to sweep the soot out of the tubes to get the best results for the coal burned. The method usually employed in sweeping the tubes is to use the *steam tube cleaner*, which consists of a length of steam hose one inch or more in diameter, one end of which is attached to any convenient steam connection, with a nozzle on the other end. The nozzle is inserted into the front end of the tube and the steam turned on, the jet of steam blowing the soot into the back connections. If the soot has become incrusted in the tubes, a *wire tube brush* or a *tube scraper* will be required to remove it. As the connection doors must be open while sweeping tubes, the operation should be performed as quickly as possible and the doors then closed. Only one nest of tubes should be swept at a time.

30. Cutting Out a Boiler.—Emergencies frequently arise whereby it is necessary to quickly cut a boiler out of service for the purpose of making temporary repairs, or to prevent damage or interference in working the fires from escaping steam or scalding water. Such casualties, for instance, as the collapsing of a tube, the blowing out of a manhole or handhole gasket, or a serious leak developing anywhere about a boiler will necessitate such a course. When it becomes necessary to cut out a disabled boiler, the fires should be quickly covered with wet ashes and the steam worked off by the engines until the pressure commences to fall; then the stop-valve of the disabled boiler should be closed and its safety valve opened. When the pressure has fallen to a point considerably below the normal, the fires may be hauled, care being taken that enough pressure is retained to blow the water level below the leak, in case it should be necessary to do so. If the leak is below the normal water level, the feed should be kept on full until after the fires are hauled so that the heating surfaces will be

protected while hauling the fires; then shut off the feed and open the bottom blow cock and blow the water level below the leak. After the boiler has cooled sufficiently proceed to repair the leak. If the boilers are fitted with dumping grates, the fires may be dropped into the ash-pits instead of being covered with ashes; or if the furnaces are provided with spray nozzles, they may, in case of a great emergency, be used for extinguishing the fires.

31. Main Feed-Pump Broken Down.—Should the main feed-pump become inoperative, one or more of the auxiliary pumps should be immediately put into operation to supply the boilers with water. The main feed-pump should then be examined to ascertain the trouble. It may be that a valve is caught up or broken; the piston or plunger packing may be blown out, or leaking to such an extent that the pump can neither take nor throw water. The feedwater may be so hot that vapor has been formed in the pump barrel and prevents the water flowing into it. If the derangement is in the valves, they should be repaired or new ones inserted. If the piston or plunger packing is the cause of the trouble, it should be renewed. If the feedwater is too hot, its temperature should be lowered by giving the condenser more injection water.

32. Relieving Watches.—The men on duty at a given time in the fireroom and engine room, that is, the watch, are changed every four or six hours, as a general rule. The watch coming on duty to relieve the outgoing watch must not only be prompt, but must also be satisfied by proper and careful examination that the fires, ash-pits, etc. are all right, that the boilers contain plenty of water, that the feed check-valves are not stuck, that the steam pressure is not unduly low, that no blow-off cock has been left partly open, that all ashes have been disposed of, etc. before relieving. Whoever is in charge of the incoming fireroom watch must bear in mind that by relieving the outgoing watch he at once assumes full and undivided responsibility for all existing conditions in his department, and cannot shift this responsibility by

the plea of lack of knowledge. He is supposed not to assume responsibility until he knows that everything is in good and proper condition, and to refuse to relieve until matters that are wrong have been righted.

COMING TO

33. As a vessel approaches her port of destination, the fires should be allowed to burn down gradually, so that they will be nearly burned out by the time the vessel is secured to the dock or at anchor. The object of this is to save coal and to obviate the waste and noise of blowing off the surplus steam through the safety valves; it also saves the labor of hauling or banking heavy fires. The exact time to stop firing depends on the kind and quality of the coal burned, the type of boilers used, and a knowledge of the harbor, or on definite information as to the time the vessel will be brought to. Even then good judgment is essential, else the fires may burn out before the vessel is secured, which might result disastrously in a harbor crowded with shipping or on encountering an unusually strong ebb tide.

34. As the fires burn down, the steam pressure should also be allowed to drop considerably below the normal, but not below the point required to work the engines properly. The object of this is to keep the steam pressure under control and obviate blowing off steam through the safety valves or opening the furnace doors and connection doors while the engines are being worked to signals. If the pressure should rise to the blowing-off point, the excess of steam may be disposed of by using the bleeder. When the signal to slow down the engines is received from deck, the blowers should be stopped, if forced draft is used, and the damper closed. If the pressure has been allowed to fall sufficiently, nothing else need be done for the time being but to stand by for the next signal from deck. This will probably be to stop the engines, followed by a series of signals to back, stop, and go-ahead, which is called *working the engines to signals*, and is for the purpose of bringing the ship to her berth. During

this time the blowers, damper, feed stop-valve, and bleeder should be operated as circumstances require, for which no set rules can be laid down, as the conditions will vary with each case. The main object is to keep the pressure under control and avoid blowing off steam or opening the furnace and connection doors if possible, but should the necessity of doing so arise there should be no hesitancy in resorting to these methods of reducing the pressure.

35. If one of the main boilers is to be used as the donkey boiler, the fires should be kept in it, and, after the ship has been secured, all communication between it and the other boilers should be closed, and the auxiliary steam and feed-systems to it should be opened. If the vessel is provided with a separate donkey boiler, the fire may be started in it with live coals from the main boilers.

36. After the vessel is secured in her berth, the notification "through with the engines" will be received from deck. The engineer of the watch will then give orders as to what shall be done with the fires—whether hauled, banked, or allowed to die out. Hauling fires is not good practice, as it allows the boilers to cool down too rapidly; the modern and best method is to let the fires die out. This is done by closing all the doors and the damper for the purpose of excluding the air from the fires. The common practice is to keep the boilers closed until the next morning after coming to, when they will be cool enough to enter for the purpose of cleaning them. While the boilers are being closed up, the boiler stop-valves should be closed, also the feed stop-valves, the steam should be shut off the steering engine, and the drain cocks on the shut-off steam pipes opened.

If the vessel is to remain in port a few hours only, the fires are usually banked. In this case, the boiler stop-valves are allowed to remain open, but the main-engine stop-valves are closed.

37. If the boilers require examination and subsequent overhauling they must be emptied by blowing down or otherwise. The method of procedure depends on the location of

the boilers, the time the vessel will be in port, and the facilities provided for emptying the boilers.

If conditions permit, it is best to allow the boilers to cool down and to pump the water from them, if located below the water-line of the vessel, or to let the water run out through the bottom blow-off cocks if the boilers are located above the water-line, as is common in river steamboats. If this practice is adopted, none of the impurities in the boilers will be baked on the plates, tubes, etc. Either a manhole or handhole above the water-line of the boilers should be opened, or the safety valves should be blocked open, to admit air to the boilers, before attempting to empty them.

When the boilers are located below the water-line of the vessel and cannot readily be pumped out, either for lack of time or facilities, they must be blown down under steam pressure. This should be allowed to fall to, say, 20 pounds per square inch, and the blow-off cocks opened gradually and not too much in order to keep down the vibrations of the hull caused by blowing the hot boiler water into the cold water surrounding the vessel. Care must be taken to close the blow-off cocks promptly when the boilers are empty.

MANAGEMENT IN PORT

CLEANING, OVERHAULING, AND LAYING UP

CLEANING AND SCALING BOILERS

38. After a vessel has arrived in port, and as soon as the boilers have cooled off, the furnace doors, ash-pit doors, and connection doors are opened, the boilers are emptied by whatever method is most convenient, and the manhole and handhole cover-plates are taken off. The tubes are then swept, and the boilers cleaned out and scaled. The scale is removed by means of scaling hammers, steel scrapers, and chisel bars. Especial care should be taken not to force the edges of the chisel bars into the seams and thereby cause leaks. All grease should be carefully removed from the water side of the heating surfaces. After all the scale has been loosened and swept into the water bottoms, it may be drawn out through the lower handholes by a small long-handled hoe. A strong stream of water from a hose directed through the upper manhole of a boiler will assist very materially in dislodging loosened scale and dirt and washing them into the water bottom. The stream of water may then be directed into the water legs and water bottoms to wash them out. If any evidence of leaks is discovered during the process of cleaning and scaling, the location of the leaks should be marked with white chalk, so that they may afterwards be readily found for calking.

After the boilers have been cleaned, scaled, and washed out, the engineer in charge of the work should make a thorough inspection of them. It is assumed that he has kept a list of all defects that have developed during the last run, and it is now his duty to closely examine those defects

and determine how they shall be treated. After entering a boiler, the engineer should note if it has been properly cleaned and scaled; he should caliper the braces to ascertain if they have been thinned by corrosion to any extent; also, he should examine the ends of the braces or stayrods to ascertain if the fastenings are in good condition, tapping the braces lightly with a hammer to learn if any are loose. He should look at the gauge and other pipe-connection openings, and at the fusible plugs, examine the leaks that were marked with chalk while the boilers were being cleaned, and any other leaks that may be on his list, look through the tubes and examine the tube ends, inspect the dry pipe, look for blisters, bulges, and cracks in the plates and, if any are found, ascertain their extent and determine how they shall be treated. If any plates are found that show extensive corrosion, they should be drilled and their thickness measured. After the engineer has made a thorough inspection of all the boilers, he should at once commence work on the repairs.

OVERHAULING

39. Overhauling boilers consists principally of calking the leaks, reexpanding or renewing leaky, corroded, or split tubes, cutting out and renewing those rivets and staybolts that leak too much to be calked effectively, replacing soft patches by hard patches, and treating cracks, bulges, and blisters in plates, if any are found, according to the necessities of each case. After the repairs in the interiors of the boilers are completed, remove from the boilers all tools, lamps, pieces of waste, etc., and clean all oil and grease off the water side of the heating surfaces very thoroughly. Paint the exteriors of the boilers, if they require it, and repair the boiler coverings. Make new joints in pipes, and renew split pipes and pipe coverings wherever needed. Overhaul all cocks and valves in the fireroom and grind in or pack all those that leak. Examine and clean out all drain pipes. Clean off all manhole and handhole cover-plates and their seats and renew all defective gaskets. Repair

bridge walls. Examine and renew all defective grate bars and bearing bars. Test steam gauges if they have shown any derangement. After all these instructions have been carried out, the boilers may be reported ready for service.

LAYING UP BOILERS

40. When a vessel is to be laid up for an indefinite length of time the boilers should receive the closest attention, as no part of the equipment of a steam vessel will deteriorate more rapidly when not in use than the boilers, situated, as they are, in the hold of the vessel where, in laid-up ships, the atmosphere is always damp. If the boilers are not properly laid up and well cared for afterwards, it is very probable that they will be found to be so much corroded when the vessel is again put into commission that they will require extensive repairs or else be entirely useless as steam generators, thus entailing great expense to repair or renew them and causing serious delay in the use of the vessel at a time when her services may be greatly needed. The tubes are particularly susceptible to deterioration when exposed to dampness. Under these conditions, they soon become pitted, and it is merely a question of time when they will become corroded through in places, which will necessitate an entire new set of tubes. The metal of the boilers in a laid-up vessel's hold is usually several degrees colder than the surrounding atmosphere, and if there is much moisture in the air, as there will be whenever the weather is damp, the boilers will sweat; that is, the moisture in the air will be condensed on the boiler plates in the form of beads or drops, which will trickle down the surfaces of the plates, leaving wet streaks, producing the best possible condition for corrosion.

41. When a vessel is about to be laid up, two points regarding the boilers must be taken into consideration: whether the vessel is to be laid up with the intention of thoroughly overhauling and repairing her before putting her into active service again, or whether she has been overhauled and repaired before being laid up, so as to be in

readiness for immediate service when needed. In the first case, the boilers need to be simply cleaned of dirt and loose scale and thoroughly dried inside and outside preparatory to being laid up. The scale that adheres to the plates need not be removed, as it will act, to a certain extent, as a protection against corrosion. But in the second case, the boilers should be thoroughly cleaned, scaled, repaired, and painted before being laid up, and greater care should be taken of them afterwards, in order that they may be in the best possible condition when the vessel is put into active service.

42. Two methods of laying up boilers are practiced; namely, the *dry method* and the *wet method*. Which method to adopt will depend on the care that will be given the boilers while they are laid up; but whichever method is adopted they must be properly cared for, meanwhile, if they are expected to escape the destructive effects of corrosion and be found in good condition when they are required for service. Laying up the boilers properly will not alone be sufficient for their best preservation; they must be kept under constant observation, and all deterioration must be checked as soon as discovered.

43. The **dry method** of laying up boilers is to thoroughly dry all parts of the boilers, particularly the water bottoms and water legs, before laying them up; and keep them perfectly dry while they are laid up. To accomplish this, all the manhole and handhole cover-plates should be taken off to afford a circulation of air through the boilers, and they should be examined at least once every day, particularly during damp weather, and if any moisture appears on the plates, light fires of shavings should be started in the furnaces and the fires kept up until all moisture has disappeared. Or, better still, one or more large coal stoves should be set up in the fireroom and fires started in them whenever any moisture appears on the plates. To carry out this method properly, the furnace, connection, and ash-pit doors and the damper should be opened and the smokestack hood should either be taken off or else raised a short distance

above the top of the pipe, so that there will be a circulation of warm air through the fire sides of the boilers. As an additional precaution, the boilers should be occasionally inspected by an experienced marine engineer and his suggestions and recommendations in regard to their preservation should be closely followed. The cost of carrying out this method of preserving the boilers of a laid-up vessel will be trifling compared with the loss by deterioration if they are neglected.

44. Another dry method of laying up a ship's boilers that is sometimes practiced is as follows: After the interiors of the boilers have been thoroughly dried, pans of chloride of lime are placed inside of the water spaces and the manhole and handhole cover-plates are put on to exclude the outside air. The chloride of lime, having a great affinity for the vapor of water, will attract and absorb the moisture from the air inside the boilers, and thus prevent it condensing on the plates. But this is only a half-way measure, so to speak, and it is not to be recommended, because the water sides of the plates only are protected, while their fire sides and the outside surfaces of the shells are left to take care of themselves.

45. The **wet method** of laying up boilers is as follows: After the boilers have been cleaned out, the manhole and handhole cover-plates are put on and screwed up water-tight; the stop-valves, and all other valves on the boilers, are tightly closed. The boilers are then entirely filled, up to the safety valves, with water, and they are thus allowed to remain until the vessel is again required for service. The idea underlying this method is that since the air is excluded from contact with the metal by the water, oxidation of the metal, that is, corrosion, is prevented. There are several very grave objections to this method, however. In the first place, the surfaces of the steam spaces and water spaces are the only parts that are protected by the water, and all the other parts of the boiler are exposed to the effects of corrosion. Moreover, it is hardly possible that a boiler, while cold, will be entirely free from small leaks through which water will find its way

and trickle down over the plates, leaving streaks of dampness in their wakes. The damp streaks will quickly corrode, and if unchecked, the process of decay will progress rapidly. If this state of affairs is permitted to continue without hindrance for a considerable length of time, the boilers will be found to be in a deplorable condition when the time arrives to put them into service, and the necessity of a general overhauling will be the result. It is, therefore, plain that, unless the boilers are entirely free of leaks, the wet method of laying them up is attended with considerable risk.

46. It seems to be the consensus of opinion among marine engineers that the dry method is the better method of laying up marine boilers, if it is properly carried out, for the reasons that all parts of the boilers are open to observation, and if any deteriorating influences arise they may be detected and checked in their early stages. In summing up the evidence on all sides, the conclusion may be reached that the preservation of laid-up boilers will depend principally on their being kept free from moisture.

INSPECTION

OCULAR INSPECTION

47. Inspection of boilers is one of the most important duties of an engineer, because on this depends largely their safety and good condition. Owing to the several deteriorating agents that tend to weaken and shorten the life of a boiler if their effects are neglected, it is important that inspections should be held periodically and notes made of the general condition from time to time. An engineer should not depend entirely on the report of a government inspector on the condition of his boiler, but should make inspections himself and actually see the condition of the boiler, the idea being that very valuable knowledge can thus be gained.

Every part, both external and internal, should be thoroughly examined. Corrosion and its progress should be noted and

action taken to entirely stop it or at least to limit it in extent. Leaks should be looked for and stopped immediately. The internal surfaces of the plates should be examined at the water-line for pitting and at the junction of the plates in the seams for grooving. The condition of stays and braces, and whether they are tight, should be noted. Should the staying be found loose, it must be tightened by whatever means are available.

HAMMER AND HYDROSTATIC TESTS

48. The ocular inspection should be accompanied by striking the plates, stays, and tubes with a hammer to determine their soundness; this is called a **hammer test.** Sound plates and tubes when struck with a hammer emit a clear, bell-like ring, while those that are thin or defective give forth a dull sound, similar to that of a piece of cracked pottery. A broken stay gives a peculiar sound that cannot be described by words.

49. The **hydrostatic test** (filling the boiler with water and applying a pressure by means of a pump or otherwise) is valuable only in showing leaks and to determine the ability of the boiler to withstand a prescribed pressure. It will not reveal weak places, unless such places are so weak as not to be able to stand the required pressure. But it frequently happens that thin places do stand the pressure to an astonishing degree, although they are in a dangerous condition; hence, the hydrostatic test should always be supplemented by an ocular inspection and a hammer test.

An objection to the hydrostatic test is that there is danger of straining the plates beyond the elastic limit and that thereby a boiler may be permanently injured which would have been safe at the working steam pressure. The inspectors in most cases depend on the hammer test and on ocular inspection, but use the hydrostatic test for new boilers, old boilers that have just been extensively repaired, and all boilers that cannot be examined thoroughly inside and outside.

When applying the hydrostatic test, the escape of air from the boiler while filling with water should be provided for, leaving some valve or cock open until the water is forced out in a solid stream. The valve or other opening used for the escape of air must be located as high as possible, so that practically little or no air remains in the boiler when it is closed. The necessity of this precaution is obvious when it is considered that, should the boiler burst under pressure while still containing air, the parts, by reason of the expansion of the air, are liable to fly with great force, perhaps injuring some one in their flight. A boiler, from which all air has escaped, bursting under the hydrostatic test will not do any serious damage.

When there are two or more boilers connected by piping, the intercommunication being broken only by a valve, it will be necessary to place a blank flange between the valve and the boiler that is to be tested, thus completely isolating the boiler from those in operation. This is done as a measure of safety, which the valve alone is not capable of insuring.

50. In making the hydrostatic test, the pressure must be applied very slowly and carefully, and the gauge must be watched for any drop of pressure that would denote a yielding of some part of the boiler. New boilers are tested by hydrostatic pressure to reveal leaky joints or rivets. When the seams or rivets are not tight, water trickles out in drops or spins out in a stream. Such places are marked with chalk and afterwards recalked.

The rules of the United States Board of Supervising Inspectors of Steam Vessels, of the British Board of Trade, and of the Canadian Inspection Service provide that the hydrostatic test pressure shall be $1\frac{1}{2}$ times the working pressure, while Lloyd's rules and those of the Bureau Veritas demand a hydrostatic test pressure of double the working pressure.

51. A method of applying the hydrostatic test that is used by many engineers is to fill the boiler full of cold water and build a gentle fire in the furnace. As the temperature

of the water rises, it expands, and thus subjects the boiler to pressure. It is urged in favor of this method that the pressure is raised steadily, and that the boiler is not so liable to injury as it is when subjected to sudden and jerky rises of pressure due to the working of a pump. The temperature of the water should in no case be made to rise above 212°, since, if a rupture should take place, the pressure of the water would lower to that of the atmosphere, and the temperature of the water being above the boiling point at atmospheric pressure, a quantity of the water might suddenly become steam and cause an explosion.

The inspection of steam boilers should begin at the place where the plates are manufactured and continue as long as the boiler is in use.

MARINE-BOILER REPAIRS

INTRODUCTION

WEAR AND TEAR

CORROSION

1. **Definitions.—Corrosion** in boilers may be defined as the eating away or wasting of the plates due to the chemical action of impure water, or due to moisture. It is probably the most destructive of the various forces that tend to shorten the life of a boiler. Corrosion is of two forms—*internal* and *external*. **Internal corrosion** may present itself as: *uniform corrosion*, *pitting* or *honeycombing*, *grooving*.

2. **Internal Corrosion.**—In cases of **uniform corrosion**, large areas of plate are attacked and eaten away. There is no sharp line of division between the corroded part and the sound plate, and oftentimes the only way of detecting the corrosion is by testing the suspected plate with a hammer and then drilling a hole through it to ascertain its thickness. Corrosion often attacks the staybolts and rivet heads.

3. **Pitting** or **honeycombing** is easily perceived. The plates are, in spots, indented with holes from $\frac{1}{32}$ to $\frac{1}{4}$ inch deep. The appearance of a pitted plate is shown in Fig. 1. On the first appearance of pitting, the affected surface should be thoroughly cleaned and a good coating of thick paint made of red lead and boiled linseed oil applied.

This treatment should be given from time to time to insure protection to the metal.

4. **Grooving** is generally caused by the buckling action of the plates when under pressure. Thus, the ordinary lap joint of a boiler distorts the shell slightly from a truly cylindrical form, and the steam pressure tends to bend the plates at the joint. This bending action is liable to start a small

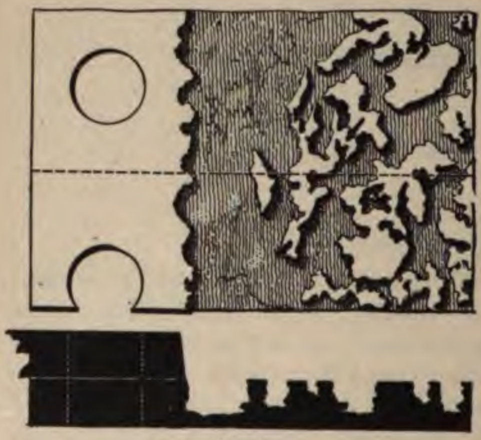


FIG. 1

FIG. 2

fracture along the lap, which, being acted on by the corrosive agents in the water, soon deepens the groove, as shown in Fig. 2. The score made along the seam by a sharp calking tool, when used by careless workmen, is almost certain to lead to grooving.

5. To prevent internal corrosion, the feedwater should be as free as possible from corrosive impurities. When bad water must be used, the corrosive impurities should be neutralized by adding alkaline substances, such as caustic soda or soda ash. Zinc is much used to arrest corrosion in marine boilers. It is believed by some that corrosion is due, in some measure, to galvanic action between the non-homogeneous portions of the iron and steel plates. By placing the plates in connection with slabs of zinc, a galvanic action

is set up between the iron and zinc, which destroys the latter and leaves the former untouched.

6. External Corrosion of Fire-Tube Boilers.—External corrosion frequently attacks marine boilers, particularly when they are neglected or laid up. The causes of external corrosion are dampness caused by leakage from joints, by moisture arising from drain pipes, blow-off pipes, damp atmosphere, etc. When leakage occurs in a joint that is hidden by the boiler covering, the plates may be corroded very seriously before being discovered. External corrosion should be prevented by keeping the boiler shell free from moisture and by repairing all leaks as soon as they appear. Joints and seams should be in positions where they may be inspected for leaks. Leakage at the seams may be caused by delivering cold feedwater against hot plates; another cause is the practice of emptying the boiler while hot and then refilling with cold water before it has cooled off. The leakage in both cases may be traced to sudden contraction of the plates due to sudden cooling. In any case, abrupt changes in the temperature of the shell should be avoided. The rush of cold air into the furnaces of a boiler when the doors are opened is a fruitful source of leakage and fracture. For this reason, a boiler should, if possible, be so constructed that none of the seams comes in contact with the fire.

7. External Corrosion of Water-Tube Boilers.—In water-tube boilers of the inclined-tube type, external corrosion principally attacks the ends of the tubes close up to the headers into which they are expanded, and especially at the back ends. This is caused by the combined action of leakage and the gases of combustion, which rapidly destroys the tubes. This corrosion usually extends to from 4 to 8 inches from the headers, and soon small pinholes appear, manifesting themselves by threadlike streams of water while under pressure. In the course of time, the tubes will leak around the expanded portion in the headers, though unless the leak is a large one its presence may not even be suspected. In this type of boilers, a small leak around a tube is difficult to

locate, unless the tube is in one of the top or bottom rows. Hence, such leaks continue for a considerable length of time, partly obscured by the accumulation of soot, until the tube becomes eaten away, as described before. When, on examination, it is found that a leak exists in a tube near the center row, though the particular tube cannot be exactly located, it is advisable to expand *all* the tubes in the immediate vicinity of the one that leaks, so as to make sure of expanding the right one. This, of course, involves more labor, but when in doubt as to the exact tube it pays to do it.

OVERHEATING AND LAMINATIONS

8. **Overheating** may be caused by low water or by incrustation. When a plate is covered by a heavy scale, the heat is not carried away by the water fast enough to prevent a rise of temperature, the plate becomes red hot and soft, and yields to the steam pressure, forming a bulge as shown at *A* in Fig. 3. If the bulge is not discovered and repaired, it will stretch until the material becomes too thin

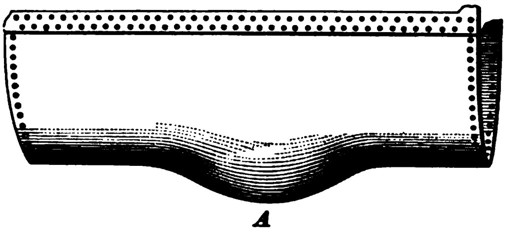


FIG. 3

to withstand the pressure, when the bulge bursts and an explosion follows. The vegetable or animal oils carried into the boiler from a surface condenser are particularly liable to cause the formation of bulges and, in Scotch boilers, the collapse of the furnaces; consequently, the greatest care should be exercised not only to keep oil out of the boilers, but also to remove frequently and thoroughly any grease that, in spite of this care, may have found its way into the boilers.

9. Laminations in a plate are sometimes developed by the action of the fire, causing a bag or blister to appear. Laminations are due to slag and other impurities in the metal, which become flattened out when the plates are rolled, as shown at *a*, *a*, Fig. 4. Under the action of heat, the part exposed to the fire will form a blister, as shown in the figure,

FIG. 4

which may finally open at the point *b* or *c*, depending on the position of the slag in the plate. The laminated portion of the plate may be very small; in that case a hard patch may be put on. If there are a number of laminations in the same plate, it is advisable to put in a new plate. When a laminated or an otherwise affected portion of a plate has to be cut out, the form of the piece cut out should be as nearly circular as possible. In any case, no sharp corners should be made, because of the tendency of cracks to start at such places.

BOILER EXPLOSIONS

CAUSES

10. Boiler explosions can be caused only by weakness of the boiler or by overpressure of steam. Either the boiler is not strong enough to carry its ordinary working pressure or else, for some reason, the pressure has risen above the usual point. A boiler may be too weak to sustain the required pressure for any of the following reasons: It may be improperly designed; the material or workmanship may be faulty; it may have become weakened by corrosion or by careless or reckless management, such as letting cold water come in contact with hot plates, or blowing off the boiler while hot, and then quickly filling it with cold water. When the pressure rises above the point for which the safety valve is supposed to be set, the fault is probably due to the

sticking fast or overweighting of the safety valve. Several very disastrous explosions have been caused by closing a stop-valve between the safety valve and the boiler while cleaning or repairing the latter, and then forgetting to open the stop-valve. The placing of a stop-valve between the boiler and the safety valve cannot be too strongly condemned; if one is there, it should be so secured that it cannot be shut.

It was formerly believed that, if the water level in a boiler fell low enough to uncover the heating surface, thus permitting the plates to become very hot, and the feedwater was then turned on, an explosion was inevitable. This hypothesis, however, has been proved by extensive experiments to be not strictly correct, but it has not yet been established under what particular or peculiar conditions a boiler will or will not explode from this cause. Therefore, the safest plan, for the present at least, is to refrain from turning on the feedwater when the exact location of the water level is unknown or uncertain.

Experiments conducted by the United States government have shown that it is possible to explode a boiler by a very sudden opening of the stop-valve. This may be accounted for thus: The sudden rush of steam from the boiler reduces the pressure materially for an instant; as the water in the boiler retains the temperature corresponding to the former pressure, part of the water flashes into steam, thus suddenly raising the pressure again and straining the boiler to the point of rupture.

PREVENTION

11. Boiler explosions may be prevented by observing the following directions:

1. Inspect the boiler frequently and thoroughly, both inside and outside. Do not rely entirely on the annual inspection by government and insurance inspectors; while this is usually quite thorough, the time elapsing between inspections of this kind is so long that the boiler may have become dangerously deteriorated long before it is time for the next inspection.

2. Use all possible care to prevent internal and external corrosion and do not permit any of the plates to become dangerously weak before renewing them.

3. Do not strain the boiler by subjecting it to too great and sudden changes of temperature; that is, do not blow it off while hot and quickly fill it up with cold water; do not deluge red-hot plates with cold water; and do not admit more cold air through the furnace doors than is absolutely necessary.

4. Do not overload the safety valves, or let them become corroded fast to their seats, or let the stems corrode fast to the bonnets.

5. Be careful not to let the water level become dangerously low.

6. Open the stop-valves slowly.

7. Do not attempt to use a worn-out boiler; replace it with a new one.

8. In selecting new boilers, adopt a type in which the danger of a disastrous explosion is reduced to a minimum.

REPAIRS AT SEA AND IN PORT

REPAIRS AT SEA

MISCELLANEOUS REPAIRS AT SEA

12. Grate bars burn out sometimes, especially when forced draft is used. When this occurs, the live coals are first cleared from the space made vacant by the burnt bar; then the blade of a slice bar is thrust into the space between the two members of a new grate bar, which is turned on its side and pushed into the furnace with the slice bar, when, by a dexterous movement, the grate bar is dropped into its place and the fire leveled.

13. The breaking of glass water-gauge tubes is of frequent occurrence, when it becomes necessary to put in new ones. When a glass tube breaks, the gauge should

be immediately shut off from the boiler; then unscrew the packing glands and remove the packing and remains of the broken tube, insert the new tube, repack the ends, and screw up the glands. Be sure that the tube is properly centered at both ends, that the glands are not screwed up too tightly, and that the shut-off valves are opened again.

14. Cracks and bulges in the internal plates of boilers are generally the result of shortness of water, deposits of scale and grease, or defective circulation. They may also be due to the weakening of the plates by corrosion and wear. Blisters are caused by laminations, which are the result of imperfect rolling. The plates that are the most liable to become cracked or bulged are those that are exposed to the most intense heat, such as the back sheets of the combustion chambers, the back tube sheets, and the crown sheets of the furnaces. These plates are also favorably located for the accumulation on them of scale or grease, and they are usually difficult of access for cleaning, all of which renders them particularly liable to become cracked or bulged. Other possible causes of plates cracking are cooling off the boiler suddenly, or admitting large volumes of cold air into it by opening the furnace and connection doors while running. Bulges that are caused by scale or grease are liable to crack; therefore, when a bulge is discovered in a boiler, that boiler should be cut out of service, the water level lowered to a point below the bulge, and the deposit of scale or grease removed; after this a careful examination of the bulge should be made. If it is found that the part that has bulged is not cracked or greatly reduced in thickness or seriously burned, the removal of the scale or grease will be all that is necessary to be done at that time, as absolutely necessary repairs only are expected to be made at sea. It is not customary, as a rule, to make any attempt to reduce a bulge while the vessel is at sea, as this would be a difficult operation to perform under the circumstances; moreover, it is hardly probable that the necessary appliances for this purpose would be found on board, whereas temporary repairs

that will enable the vessel to reach port can usually be made. Should the bulged part be considerably reduced in thickness, a staybolt or brace, reaching to and connected with the most convenient point on the shell, should be fitted at the center of the bulge. If the metal is cracked or badly burned, it should be covered with a soft patch.

15. A **soft patch** consists of a piece of boiler plate large enough to cover the defective portion of the plate to be patched, allowing sufficient margin to insure enough solid metal around the defect for the bolt holes. A templet of the patch is first made of a piece of sheet lead thick enough to hold its shape while being handled, which is cut out to approximately the size and shape the patch is to be; it is then fitted by trimming and bending to the place where the patch is to go. If the templet is very crooked or uneven after being fitted, it will have to be flattened out again for the purpose of marking off the shape of the patch on the plate from it. The templet is then again fitted and bent to the form the patch is to be, and the patch is cut out and bent, at the forge, to conform exactly to the shape of the templet. A thin shallow lip, not more than $\frac{1}{4}$ inch in depth, is turned inwards all around the patch, after which the bolt holes are drilled, or punched if it is a hurried job. The bolt holes are then marked off on the defective plate from the patch, and the holes drilled. A stiff putty, composed of red lead, white lead, and some fine cast-iron borings well mixed together is now made and applied to the inside surface of the patch to a thickness a little greater than the depth of the lip. The patch is then bolted to its place by through bolts and nuts, using washers and grommets. A **grommet** consists of a piece of cotton lamp wicking, 10 or 12 inches long, saturated and covered with moist white lead. This strand of wicking is wound tightly around the bolt, one piece under the head and another under the nut. When the nut is screwed up tight, the grommets are squeezed into the crevices between the washers and the bolt, sealing the joints, and rendering them waterproof. The excess of white lead that

has been squeezed out during the process of screwing up the nuts is then scraped off, and the lip of the patch is calked up tight against the boiler plate. In places where through bolts and nuts cannot be used, tap bolts must be substituted for them. In that case, the holes in the boiler plate must be drilled smaller than the holes in the patch, to permit tapping.

A soft patch is a temporary expedient only, but it is usually sufficient to enable the vessel to reach port, when more permanent repairs should be made.

16. There are three methods of treating simple cracks in boiler plates; which method to use depends on the extent of the crack. A crack may be closed by *calking*, by a row of *screw rivets*, or by *patching*.

If the crack is a small one, calking will generally stop the leak; the end of the crack should be first drilled out and a screw rivet put in to prevent the crack extending farther into the plate.

A **screw rivet** is made by screwing a threaded piece of iron rod, slightly longer than the plate is thick, into a threaded hole in the plate and then riveting over both ends. If calking will not close the crack, a row of screw rivets may be put in, close together, throughout the entire length of the crack, and their ends calked over each other; this method will usually be effective. But if the crack is too extensive, either in length or in breadth of opening, to be closed by either of these methods, it should be soft-patched. In all cases in which cracks occur in the plates forming the heating surfaces of a boiler, the inside of the plate should be thoroughly cleared of all scale, grease, or sediment in order to prevent further development of the crack.

17. Cracks in tube sheets usually extend from one tube hole to an adjacent tube hole. If calking will not close the crack, it should be patched. The customary way to do this is to fit a patch around the holes at each end of the crack and secure the patch to the tube sheet by square-headed tap bolts, and afterwards calk around the edge of the patch. If the crack extends through several tube holes, the patch

should, of course, be made large enough to entirely cover the crack. Cracks are sometimes, but not often, found in the shell plates. The treatment of such cracks may be similar to that given to cracks in the internal sheets.

18. When blisters are discovered on the internal plates of a boiler, they should be carefully examined by sounding with a hammer, or by chipping and drilling to ascertain their extent and thickness. The blistered portion should be cut away, in most cases, as corrosion may be in progress on the internal surfaces. If a blister is thin and of small area, all that will be necessary to do for the time being is to remove it by chipping; but if it is very deep and extends over a large area, the plate containing it should be additionally stayed or braced after the blister has been removed; or else the pressure should be reduced until the vessel reaches port, when a more permanent repair should be made.

REPAIR OF ORDINARY LEAKS

19. Leaky Boiler Tubes.—In old fire-tube boilers, especially, leaking of boiler tubes is a very common occurrence. The leaks may be found around the ends of tubes that have not been properly expanded in the tube sheet, or that have become thin by frequent reexpanding. The bead at the rear end of a tube may be burned or corroded off and cause a leak. The tubes may be split or cracked at an imperfect weld or, after long use or exposure to moisture while not in use, they may become pitted by corrosion, which will ultimately cause numerous holes to develop in them. When a leak occurs around the end of a tube, if the tube is sound, the leak can usually be stopped by reexpanding the tube; but if the bead is burned or corroded off, it will be necessary to drive the tube back slightly from the front connection end and then reexpand and rebead the rear end. If the tube is too short to admit of being driven back, the leak may be stopped by driving a cap ferrule, specially made for this purpose, into the end of the tube. This ferrule is illustrated in Fig. 5, in which *a* is the tube, *b*, *b* are the tube sheets, and *c*

is the ferrule. This ferrule is known as the **Admiralty pattern cap ferrule** and is extensively used in the British navy.

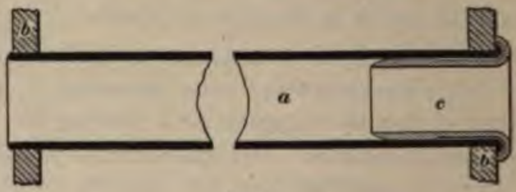


FIG. 5

20. Leaks arising from split or corroded tubes are stopped by a device called a **tube stopper,** of which several types are in common use. One of the most simple is illustrated in Fig. 6, in which *a* is the tube, *b*, *b* are the tube sheets, and *c*, *c* are cast-iron plugs with cupped flanges, one plug being placed in each end of the leaky tube. These plugs

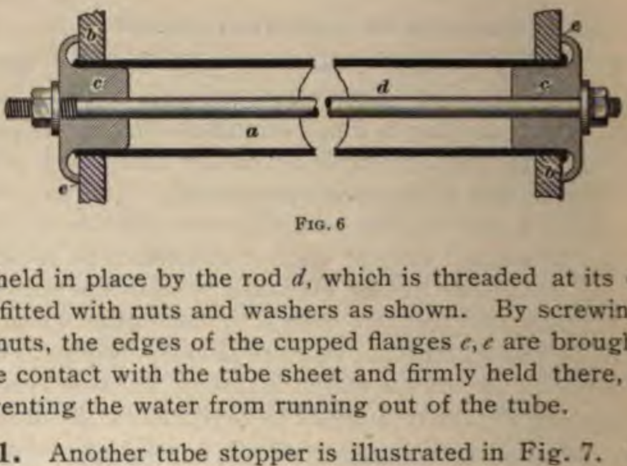


FIG. 6

are held in place by the rod *d*, which is threaded at its ends and fitted with nuts and washers as shown. By screwing up the nuts, the edges of the cupped flanges *e*, *e* are brought in close contact with the tube sheet and firmly held there, thus preventing the water from running out of the tube.

21. Another tube stopper is illustrated in Fig. 7. This stopper is similar to the one shown in Fig. 6, except that the cupped washers *a*, *a* are substituted for the cast-iron plugs *c*, *c*, Fig. 6. The stoppers shown in Figs. 6 and 7 will answer the double purpose of stopping leaks at the tube ends as well as leaks inside the tubes, but they put the tube out of service in either case.

22. Yet another form of tube stopper that is used to stop leaks inside of tubes is illustrated in Fig. 8. The advantage of this stopper is that it can be inserted into a tube without

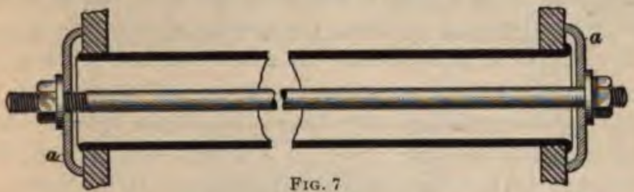


Fig. 7

having to draw the fires. Referring to Fig. 8, it will be seen that the washers *A*, *B*, having the rubber ring *f* between them, are placed on each end of the tie-rod *C*. This rod is threaded at both ends and fitted with nuts and washers. The sleeve *D*, made of ordinary steam pipe, is placed between the two pair of washers. The distance *L* is about 1 inch less than the length of the tube. To stop a leak in a tube with this stopper,

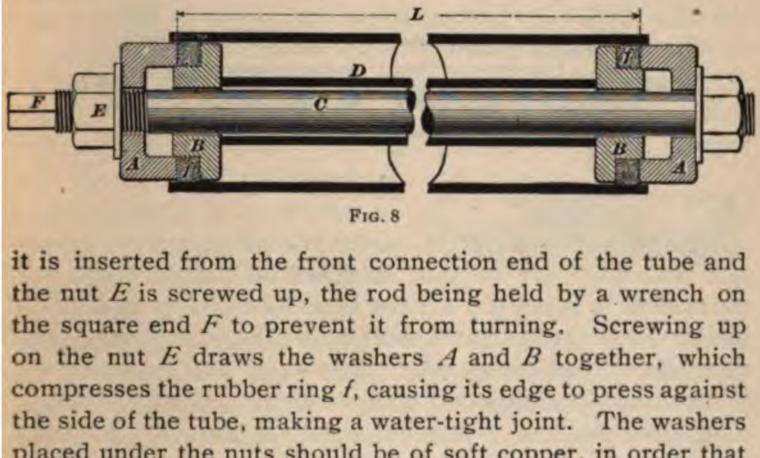


Fig. 8

it is inserted from the front connection end of the tube and the nut *E* is screwed up, the rod being held by a wrench on the square end *F* to prevent it from turning. Screwing up on the nut *E* draws the washers *A* and *B* together, which compresses the rubber ring *f*, causing its edge to press against the side of the tube, making a water-tight joint. The washers placed under the nuts should be of soft copper, in order that a water-tight joint may be made.

23. Another tube stopper of the same class as that illustrated in Fig. 8 is shown in Fig. 9. It consists of an iron tie-rod *a*, threaded at both ends and fitted with nuts and washers. Four large metal washers *b*, *b*, just slightly less in

diameter than the inside of the tube, are fitted on the rod in pairs; between each pair of these washers are the thick rubber washers *c*, *c*, which are made to closely fit both the tube and the rod. The sleeve *d*, made of ordinary steam piping, is for the purpose of holding the group of washers at each end of the rod in place while the nut *e* is being screwed up, the rod being held by a wrench put on the square end *f* of the rod to prevent it from turning. To stop a leak inside of a tube with this stopper, the nut *e* is first slacked off, then the stopper is inserted into the front end of the tube and pushed into its place; the nut *e* is then screwed up tight; this compresses the rubber washers and squeezes their edges against the inside of the tube and also against the rod, making water-tight joints that stop the leak very effectually. This stopper is to be preferred to the one shown in Fig. 8; moreover, it can be made on board ship from materials usually carried in store.

FIG. 9

24. When a tube gives out from either a split or corrosion and there are no tube stoppers on board, or it is not convenient to make the stopper described in Fig. 9, wooden plugs, made preferably of white pine, may be driven into each end of the disabled tube. The water will cause the plug to swell and the tube will fill up with salt, which will effectually stop the leak. With this method, however, it will be necessary to haul the fires and blow off the pressure from the boiler containing the leaky tube. A very simple and effective method of stopping an internal leak in a tube is to drive a snugly fitting plug of soft wood into the leaky tube from the front end, forcing the plug back beyond the leak, and then to close up the front end of the tube with another wooden

plug; or, for a boiler carrying high pressure, take a soft-wood plug, cut it down in the middle as shown in Fig. 10, and drive it into the tube so that the leak *a* will be opposite the smaller portion of the plug. The pressure cannot blow out this tube stopper as it might do if separate plugs are used.

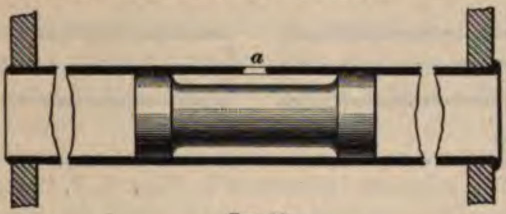


FIG. 10

25. Another method of plugging a leaky tube is by introducing a split sleeve or ferrule made as shown in Fig. 11, which also shows its method of application. The sleeve may be made of a piece of old boiler tube and may be given a length of from 1½ to 2 diameters. The piece of tube is split lengthwise into two pieces, which are bent slightly to conform to the inside of the leaky tube. The piece should be split at an inclination of about ⅜ inch to the foot and the edges filed smooth and true. After smearing the inside of the leaky tube with red-lead putty, one half of the sleeve is put in flush with the tube end; the other half is placed on top, as shown in Fig. 11, and then driven home flush with the end of the tube. By reason of the edges being in a plane inclined to the axis, the two halves of the sleeve are firmly wedged against the inside of the leaky tube when the second half is driven home. The sleeves must be neatly fitted; in that condition they have been used with good success, effectually stopping a leak and obviating the necessity of withdrawing the tube or losing its service.

FIG. 11

26. An easily made tube stopper is illustrated in Fig. 12. It consists of two tapering pine plugs having a central hole through which a rod, made of ½-inch round iron, is passed. This rod is provided with nuts and washers at each end, by

means of which the plugs can be drawn home. This tube stopper is quite effectual so far as stopping the leak is concerned, but it is open to the objection that a man must enter the back connection in order to place the plug in the rear end in position and in order to put the nut and washer on the rod.

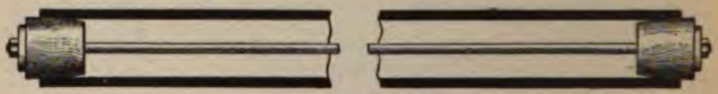


FIG. 12

27. The stoppers illustrated in Figs. 8, 9, 10, and 11, and the wooden plugs driven into a tube from its front end, can be inserted without hauling the fires, but they can be used only when the tubes are reasonably free from hardened soot and salt. When a tube is nearly filled with incrustation, as leaky tubes are very liable to be, it will be impossible to insert the stoppers into a tube until the incrustation is removed. This is not an easy task, as the mixture of salt and soot in the tube will be baked about as hard as granite by the intense heat to which it is exposed, and a steel chisel bar will be required to remove it; an ordinary tube scraper will have no effect on it. Now, if the tube is in an advanced stage of corrosion, the blows of the chisel bar on the hard incrustation will be very liable to break more holes in the tube, and probably almost wholly destroy it; therefore, it is not always advisable to undertake to clear out a choked tube; this should never be attempted with pressure on the boiler, for the man handling the chisel bar may be badly scalded. When a tube is choked with incrustation, it is preferable to use either of the stoppers illustrated in Figs. 6 and 7, or else clean the scale out of the ends of the tube and drive in wooden plugs.

28. To insert the Admiralty ferrule, or the stoppers illustrated in Figs. 6, 7, and 12, into a tube, it will be necessary to first haul the fire from the boiler having the leaky tube, close the stop-valve, and blow off the pressure. It may also be necessary, if the leak is a large one, to run the water level

below the leaky tube to enable a man to safely go into the back connections to adjust the rear end of the stopper or to insert the ferrule. Although cases are known in which men have performed this operation while the pressure was on the boiler, it is a dangerous practice and is not recommended.

29. The stoppers illustrated in Figs. 7, 9, 10, 11, and 12 have the merit of being easily made on board ship of materials usually carried in store. The rubber washers *c, c*, Fig. 9, should be made of pure gum; an old, soft-rubber, air-pump or circulating-pump valve will answer very well for this purpose.

The Admiralty ferrule will stop a leak at the ends of a tube only. The stoppers illustrated in Figs. 6 and 7 will stop leaks both inside and around the ends of the tubes, while the stoppers illustrated in Figs. 8 to 12, and the wooden plugs mentioned, will stop internal leaks only. It will thus be seen that each class of stoppers has its own sphere of usefulness, and in fitting out an engine department of a vessel some of each class, or materials to make them, should be provided. The dimensions of the ferrules or stoppers and their different members will, of course, depend on the sizes of the tubes in which they are to be used. There are a number of other tube stoppers and ferrules in use, but those described above are examples of the representative types.

30. Plugging leaky tubes is only a temporary expedient, and, therefore, as a matter of course, those plugged tubes that are too far gone to be reexpanded should be cut out and new ones inserted on the vessel's arrival in port. When several tubes give out on account of corrosion, it is strong evidence that all the other tubes are in very nearly the same condition, and the safest plan, in that case, will be to entirely retube the boiler while in port. When tubes begin to give out from corrosion, they are apt to go one after another in rapid succession, and if a vessel goes to sea in that condition she may become disabled at a critical time. The condition of those tubes that are cut out will be a standard to judge the others by; if they are in an advanced

stage of corrosion, all the others are probably in a similar condition.

31. Leaky Rivets, Staybolts, and Seams.—Leaks frequently occur at rivets, staybolts, and seams, but they are usually very small at first and do not require any immediate treatment. Small trickling leaks will frequently **salt up** in time; that is, the water that oozes through the leak will be evaporated, leaving such impurities as it contains in a hard mass around and over the leak, which closes it effectually. If, however, the water squirts out of the leak in jets, it is not probable that the leak will salt up; on the contrary, it is apt to increase. Leaks of this kind should be closely watched, and if they increase to such an extent, before reaching port, that they interfere with the fires, they should be repaired. This will necessitate cutting out of service the boiler containing the leak and allowing it to cool down sufficiently to work in. Usually, leaks of this character can be stopped temporarily by calking, but in the cases of leaky rivets and staybolts, especially if they have been calked several times before, the leak cannot always be stopped in this manner. It will then be necessary to cut out the defective rivet or staybolt and insert bolts and nuts in their places, or tap bolts, if the location is inaccessible for putting in a bolt and nut. Extra large cupped washers should be put under the heads and nuts of these bolts, the cupped part of the washer being filled with a stiff putty composed of a mixture of white and red lead and a small quantity of fine cast-iron borings. To further insure against leakage around the bolt, grommets should be placed under the head of the bolt and under the nut.

32. If the leaky staybolt is of the socket type, either the new bolt or a mandrel should be pushed into the hole and through the socket as the old bolt is being withdrawn, in order to prevent the socket, in case it should happen to be loose, dropping into the water bottom of the boiler, from whence it might be troublesome to recover it. If the staybolt to be cut out is of the screw type, and it is desired or

necessary to replace it with a socket bolt, and if its location in the boiler is such that it cannot be reached by the arm or tongs, a very good plan to get the socket into its place is to pass a string through both holes and secure the ends, dropping the center and hauling the bight through a handhole; then cut the string, pass one of the ends through the socket, join the ends of the string together again, and haul the socket to its place. In fitting sockets, it is important that their lengths should equal the exact distance between the sheets, and that the ends should be filed square, otherwise the sheets may be drawn out of shape.

33. Leaks in seams at the junction of three plates are often very troublesome to stop. No amount of calking, seemingly, will close them. Sometimes, a small steel pin or wedge may be driven into the leak and the plate may be calked over the pin or wedge; this will be effective in stopping such a leak.

34. Leaky Manholes and Handholes.—When leaks occur at manholes and handholes, they should be stopped at once, or the gasket may be blown out and the boiler temporarily disabled. If the gasket is sound and the leak is attended to as soon as discovered, it may be stopped by screwing up the nuts on the studs or bolts of the plate. This is an operation that requires care and should be performed only by a man experienced in that kind of work, as otherwise a stud might be twisted off, which would be apt to let the plate spring away from its seat sufficiently to allow the gasket to be blown out with disastrous results. The nuts should not be screwed up beyond hand taut with a moderate-sized wrench; a sledge hammer should never be used to drive up the wrench in a case of this kind. Should a gasket be blown out, the only course to pursue is to cut out that boiler, blow the water level below the leak, and make a new joint with a new gasket. A supply of spare gaskets should always be carried for such emergencies.

35. Leaky Blow-Off Cocks.—Should it be found difficult to maintain the water level at its proper height in any

of the boilers when the feed is on full, it is strong evidence that some of the water is leaking out of the boiler. After establishing the facts that the proper amount of water is going into the boiler, and that the drain cock is closed, both the surface and bottom blow cocks should be examined to ascertain if they leak or are partly open. This inspection may be made by feeling the pipes outside the cocks with the hand. If the pipe is quite hot, it indicates that water from the boiler is leaking through the cock, or that the cock is not entirely closed, or that the plug of the cock is slack or considerably worn. It is plain what should be done in the first two instances; namely, entirely close the cock or tighten the plug. In the other case, close the outboard valve of the blow-off pipe and keep it closed while the blow-off cocks are not in use, and tighten up or grind in the plug at the first opportunity. If the plug is too much worn to be ground in successfully, put in a new cock.

36. Leaky Pipes.—Both steam pipes and water pipes are liable to leak from several causes, one of the most common being defective joints. Joints in pipes are of two kinds; namely, screw joints in small pipes and flanged joints in large pipes. Other sources of leaks in pipes are cracks and pinholes. A pinhole in a steam pipe will increase in size very rapidly, as the jet of steam issuing therefrom will cut away the metal around the hole considerably in a very short time. In the cases of leaks occurring at a screw joint or on account of a crack or pinhole, temporary repairs can be made by wrapping the pipe at the leak, and for some distance each side of it, with a strip of coarse canvas with a thin layer of white lead spread on it. After the canvas has been wrapped tightly around the pipe, it is secured by being sewed with marline or annealed wire, which is wound around the pipe in close coils, hauled taut, and the end securely stopped. Iron clamps may also be used for this purpose, if they are at hand. In the case of a leak occurring in a flanged joint, it can usually be stopped by setting up on the joint bolts, provided that this is done in time; but if the leak is permitted

to continue for some time, the chances are that the gasket will be blown out, in which event it will be necessary to cut that pipe out of service and make a new joint with a new gasket. Should a gasket be blown out of a joint in the main steam pipe, it will necessitate shutting down the main engines for a considerable length of time to make repairs. As it is not an easy matter to make a new joint in a large pipe at sea, especially in rough weather, the importance of promptly stopping a leak in a joint in the main steam pipe is apparent.

REPAIRS IN PORT

INTRODUCTORY REMARKS

37. After a steam vessel has been in active service for some time, the boilers will require general, and sometimes extensive repairs. Boilers that have been properly cared for, that is, kept clean and free from corrosion and not heated up and cooled down suddenly, will run much longer without extensive repairs than if these precautions had not been taken. When general repairs are to be made on boilers, the services of experienced boilermakers and the facilities of a boiler shop are required; hence, when a vessel is to be laid up for such repairs, she must be taken to a port where these facilities may be obtained. This work is usually let by contract to the lowest bidder, if there is competition, or if the company to which the vessel belongs has no shop in the vicinity; therefore, it will be necessary for the engineers of the vessel to carefully inspect the boilers to ascertain what repairs are required, and to make a list of them for the competing boilermakers to bid on.

The lesser repairs, such, for example, as expanding tubes, cutting out and putting in tubes and sleeves, calking leaky seams, rivets, staybolts, etc., may be done by the fireroom force. But when the cutting out of sheets or the portions of sheets to be patched, the driving of rivets, and similar work are to be undertaken, it is generally advisable that none but boilermakers should attempt it.

PLATE REPAIRS

38. When a defect in a boiler plate necessitates hard patching, the defective part is cut out and a templet of the patch is made of sheet lead and fitted over the cut-out part, taking care to make the templet large enough to leave sufficient lap for the rivets. The templet is then taken to the boiler shop and a boiler-plate patch of the exact size and shape of the lead templet is made, and the rivet holes are drilled. The patch is then taken on board ship and the rivet holes marked off from the patch on the plate where it is to be secured, and the holes are drilled. The patch is next held in place by a few bolts and nuts, after which it is riveted on. If the location of the patch is such that rivets cannot be driven, bolts and nuts or tap bolts must be used instead of rivets. After the patch is secured, it should be calked, both around the edge of the patch and around the edge of the hole. Patches should be put on the pressure side of the plates wherever their locations are accessible for that purpose.

39. When an entire plate is condemned, it should be cut out and a new one inserted. To remove the defective plate, the rivet heads are cut off by means of a boilermaker's cold chisel and a sledge hammer; the rivets are then backed out by a drift. After the plate is released from the surrounding plates it is taken to the shop, where a new plate is made the exact shape of it and the rivet holes are marked off and drilled; the new plate is then taken on board the vessel and riveted in its place and its edges calked. If any staybolt holes are required in the new plate, care should be exercised in marking them off, so that they will come fair with the holes in the opposite plate. The same care should be observed in marking off the rivet holes.

40. If a bulge is quite shallow, that is, not more than, say, 2 inches in depth, and the metal is neither burned or cracked, it may be reduced by first heating it in a portable furnace and then forcing it back with a hydraulic jack. But

if the bulge is of greater depth than about 2 inches, or if the metal of the bulge is stretched, it cannot be put back by that means. In this case, it must be driven back by round-faced sledge hammers. It is necessary to apply forced draft to the portable furnace to produce heat enough to drive the bulge back with hammers or force it back with a jack. If a bulge is cracked or badly burned, it should be cut out and the plate hard-patched.

TUBE REPAIRS

41. In cutting out old fire-tube-boiler tubes, the rear ends of the tubes are split for about 1 or 2 inches with a narrow-edge cape chisel, after which the ends are bent inwards toward

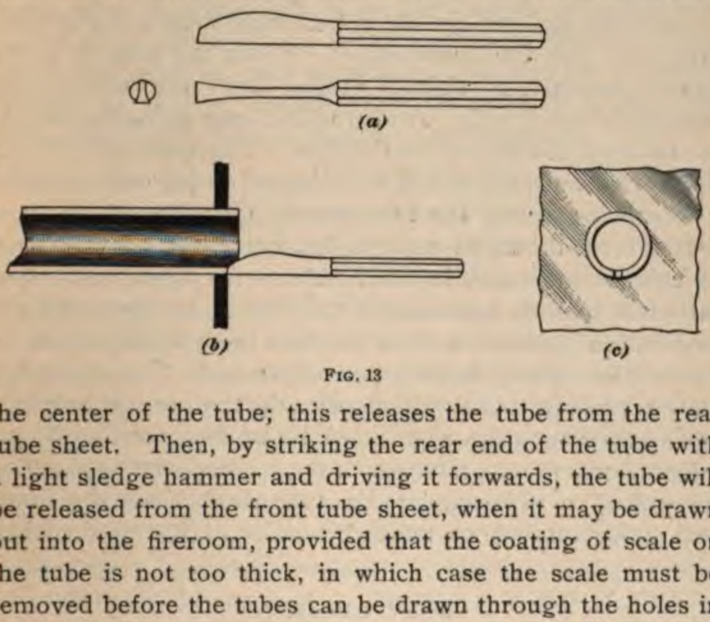


FIG. 13

the center of the tube; this releases the tube from the rear tube sheet. Then, by striking the rear end of the tube with a light sledge hammer and driving it forwards, the tube will be released from the front tube sheet, when it may be drawn out into the fireroom, provided that the coating of scale on the tube is not too thick, in which case the scale must be removed before the tubes can be drawn through the holes in the front sheet; or the tubes must be cut in two inside the boilers and the pieces taken out through the manhole.

42. A tool called a **ripper**, and shown in Fig. 13 (*a*), is sometimes used instead of a cape chisel for cutting out old boiler tubes. Fig. 13 (*b*) shows how it is applied and

Fig. 13 (*c*) shows an end view of the tube after it has been cut. By means of the ripper, a slit about $\frac{1}{8}$ inch wide and extending about 1 inch beyond the inside of the sheet is cut; the end of the tube can then readily be squeezed together so that the tube will pass through the hole. With reasonable care, there is little danger of cutting into the tube sheet. In splitting the end of a tube, care should be exercised not to turn a chip of the metal down inside the tube sheet, or the tube will be very troublesome to remove.

43. The ends of new boiler tubes should be **annealed**, that is, softened by heating them to a cherry red and allowing them to cool slowly, before being expanded. The length of the tubes should be about $\frac{3}{4}$ inch greater than the distance between the rear surface of the back tube sheet and the front surface of the front tube sheet. When the tube is in its proper position, it should project into the combustion chamber about $\frac{1}{4}$ inch. A man with a tube expander should be stationed at each end of the tube. The expanders should be inserted into each end of the tube and rolled until the tube is firmly secured in the tube sheet. Care should be exercised to roll the tubes enough, but not too much, less they be split or otherwise injured. After the tubes have been expanded into the tube sheets, the ends that project into the combustion chamber are turned over with a peening hammer and afterwards beaded with a boot tool. The stay-tubes are extra heavy, usually about twice the thickness of ordinary tubes; their ends are reenforced by welding ferrules 3 or 4 inches long to them, after which the ends are threaded. The stay-tubes are screwed into the tube sheets from the front, nuts being screwed on the ends outside of the tube sheets.

STAY REPAIRS

44. There is only one sure remedy for a leaky staybolt or rivet, and that is to cut it out and put in a new one. Should the holes of the defective rivets or staybolts be much enlarged by corrosion, they should be reamed out until solid metal is reached and rivets or staybolts of larger sizes than

the original ones put in. If a large number of new and larger rivets are put into a joint, there is a strong presumption that the safe working strength of the joint is reduced thereby; if this should be the case, the working pressure on the boiler will have to be reduced below the former pressure allowed.

Should any of the stayrods, staybolts, gusset stays, palm stays, etc. show a considerable reduction in size or be found broken, they should be removed and new ones substituted, taking care that these new stays are made of the correct length.

FURNACE REPAIRS

45. Dead plates being constructed of cast iron and exposed to extreme heat and rough usage, it is probable that some of them will be found cracked or broken. These should be renewed, as a general rule, as they are extremely difficult to repair. Furthermore, the cost of a new dead plate will usually be less than the cost of repairs.

46. It is to be expected that the bridge walls will be found in a rather dilapidated condition. Many of the bricks will probably be missing, while others will be loose or broken. All such should be removed, and only those that are held firmly in their places should be retained. Whatever remains of the wall after removing the loose and broken bricks should be thoroughly cleaned off and sprinkled with water before the new bricks are laid. Firebricks only should be used in building or repairing bridge walls; common bricks will not answer for this purpose, as they are unable to stand the heat to which they are exposed. Fireclay, mixed quite thin with water, should be used as the mortar for bridge walls. Each brick should be dipped in water just before laying it to cause the mortar to adhere to it. The bricks should be carefully laid with their edges flush on the sides and top of the walls, and then plastered over with fireclay.

47. Grate bars and bearing bars are exposed to great heat and rough usage; consequently, their life is comparatively short and they require frequent renewal. While general

repairs are going on, these bars should be overhauled; that is, the good bars sorted out from the bad and the latter sent ashore. The bearing-bar brackets should also be examined and any defects found in them remedied. The supply of spare grate and bearing bars should now be brought up to its standard quantity of one-eighth of a complete set for all furnaces, for a sea-going vessel.

48. Furnace fronts, being made of cast iron and exposed to considerable changes of temperature, are liable to crack. If any such cracked fronts exist, they should be removed. The linings of the furnace doors will probably be burnt, cracked, and warped; these should be removed. Furnace doors frequently become sagged so that they cannot be closed tightly; all such should be straightened and put in good condition.

MISCELLANEOUS REPAIRS IN PORT

49. It is to be expected that the zinc boiler protectors will require renewal occasionally. The straps for holding the zinc plates should be filed bright where they come in contact with the zinc and with the boiler braces. After the straps are bolted in place, the joints should be made water-tight by cement. There should be 1 square foot of exposed zinc surface, exclusive of edges, for each 130 square feet of heating surface in a boiler. In the British navy, zinc slabs 12 inches by 6 inches and $\frac{1}{2}$ inch thick are attached to the boiler braces; there being one slab to every 20 horsepower. These are eaten up and renewed, when the boilers are in use, every 60 to 90 days. The baskets or troughs for catching the disintegrated zinc should be examined and, if needed, put in good order.

50. Rain and sulphurous gases from the coal have a destructive effect on the metal of the uptakes and smokestack; therefore, in course of time, they will become corroded to such an extent that they must be renewed. Those sheets that are most exposed to dampness and corrosion will be the first to give out. These should be examined and renewed, if necessary.

51. Before painting the boilers, their shells should be thoroughly cleared of rust, flakes of old paint, and other foreign substances that may be adhering to them. A good paint is a mixture of red lead and boiled linseed oil. The bottoms of the boilers and those parts that are not protected by the coverings should receive two coats of paint.

After the paint on the shelves of the boilers is thoroughly dry, the coverings should be replaced. So much of the old covering as may be in good condition should be utilized and the deficiency be made up with new material. If the coverings of any of the steam pipes were removed or injured while repairing, they should now be replaced or repaired.

52. Boiler fittings are subjected to corrosion and wear and they are also exposed to accidents; therefore, they require overhauling occasionally, and the proper time to put them in order is while the ship is laid up for repairs.

53. When a vessel is laid up for repairs, advantage is taken of the opportunity to put her into dry dock and clean and paint the hull below the water-line. This gives the engineers a chance to examine and overhaul the outboard valves and strainers, and it is very important that this opportunity should not be neglected. The Kingston valves, the outboard delivery valve, the outboard blow valves, and the injection valve should be examined and put in perfect order. The flanges of valves that are secured directly to the outer hull plating should be bolted to strengthening rings by steel studs with composition nuts, care having been taken not to drill the stud holes entirely through the rings. A zinc protecting ring is fitted in each opening in the outer skin in such a manner as to be easily renewed. In vessels with double bottoms, all sea valves over the double bottoms are inside the inner skin and are connected to the outer skin by a pipe. The valve chamber is bolted to a flange on the upper end of the pipe, there being a zinc protecting ring near the upper end of the pipe, or secured to the lower flange of the valve chamber. These rings should be made accessible for renewal.

54. All copper suction and discharge pipes designed to convey salt water should be fitted, at intervals of from 10 to 15 feet, with cast composition boxes, with flanges matching flanges on pipes. These boxes are placed on horizontal sections of the pipes. The zinc protecting rings on the sea suction valves may be regarded as one of the zinc boxes in spacing the remainder. Each pump discharge pipe designed to convey salt water should have a zinc box as near the pump as is practicable, after which they should be spaced as specified above. The design of the zinc boxes should be such as to leave the pipe unobstructed. The box should have the general form of the body of a gate valve, there being a bonnet and a faced opening for inserting the zinc, which should be of **U** shape, bent from a rolled slab $\frac{1}{2}$ inch thick.

The perforations in the strainers over the pipe openings in the hull plating should be cleaned out and the strainer fastenings examined, and renewed if necessary.

MARINE-BOILER INSPECTION

(PART 1)

AMERICAN, BRITISH, AND CANADIAN RULES

SPECIFICATIONS FOR MATERIALS

INTRODUCTION

1. Practically all civilized countries have passed laws providing not only for the licensing of marine engineers, but also for the inspection of steam vessels and their machinery. In the United States of America, the inspection of steam vessels and licensing of engineers is performed by The Steamboat-Inspection Service, Department of Commerce and Labor, with headquarters at Washington, District of Columbia. In Great Britain, The Imperial Board of Trade performs a simiiliar function; in Canada, The Department of Marine and Fisheries, through its Board of Steamboat Inspection, inspects steam vessels and examines marine engineers. These various bodies are governed in their action by rules and regulations, which of course are amended and added to from time to time. Obviously, marine boilers, in order to pass inspection, must conform to the rules and regulations in force in the country in which they are built. In addition, various marine underwriters, such as Lloyd's and the Bureau Veritas, have their own rules and regulations, which must be complied with if the boilers are to be insured in these societies.

The title of the American regulations is "General Rules and Regulations Prescribed by The Board of Supervising Inspectors, Steamboat-Inspection Service"; they will be referred to hereafter as "American rules" for short. These can be obtained by applying to the Supervising Inspector-General, Steamboat-Inspection Service, Department of Commerce and Labor, Washington, District of Columbia. The British Imperial Board of Trade rules are entitled "Regulations and Suggestions as to the Survey of the Hull, Equipments, and Machinery of Steam Ships Carrying Passengers"; their price is six pence, and they can be purchased from Wyman & Sons, Ltd., Fetter Lane, E. C., London, England, or Oliver & Boyd, Edinburg, Scotland, or E. Ponsonby, 116 Grafton Street, Dublin, Ireland. The Canadian rules, which are very nearly the same as the British Board of Trade rules, can be obtained by applying to the Minister of Marine and Fisheries, Ottawa, Canada; their title is "Rules for the Inspection of Steamboats and for the Examination of Engineers of Steamboats."

2. The Canadian rules are divided into two parts, each covering virtually the same ground. Part I is normally used by the Canadian inspectors; Part II is headed thus: Regulations governing the inspection and testing of boilers now in existence and of boilers now or hereafter to be manufactured, in Canada, for the use of steamboats, whenever in the opinion of the Inspector the regulations contained in Part One of these Regulations, on account of the make of such boilers, or for some other reason, are not capable of application in the testing thereof; provided that in every such case the Inspector shall issue his certificate, in which he shall state that his inspection has been made under Part Two of this Order.

3. The American rules here quoted are taken from the edition of August 8, 1906; the Canadian rules are from the 1904 edition; and Board of Trade rules, from the 1905 edition; these editions in every case were the latest at the date of writing.

SPECIFICATIONS AND TESTS FOR WROUGHT IRON AND STEEL

4. Plates.—The American rules provide that every iron or steel plate intended for a marine boiler must be stamped with the tensile stress per square inch of section it will bear; the tenacity must not be less than 45,000 pounds per square inch for iron plate and 50,000 pounds per square inch for steel plate.

5. Steel plates, under the American rules, must be made by the basic or acid open-hearth processes, if intended for use in boilers; plates used for making tubes may be made by the Bessemer process.

Plates made by the basic process must not contain more than .04 per cent. of phosphorus and .04 per cent. of sulphur; when made by the acid process, they must not exceed .06 per cent. of phosphorus and .04 per cent. of sulphur.

The tested sample must have a tensile strength of not less than 50,000 nor more than 70,000 pounds per square inch; it must show an elongation of at least 25 per cent. in a length of 2 inches for a thickness up to $\frac{1}{4}$ inch, inclusive; for plates between $\frac{1}{4}$ and $\frac{7}{16}$ inch, inclusive, the length is 4 inches; for plates over $\frac{7}{16}$ inch, the length is 6 inches. The sample must also show a reduction of sectional area as follows: At least 50 per cent. for a thickness up to $\frac{1}{2}$ inch, inclusive; 45 per cent. for a thickness over $\frac{1}{2}$ and up to $\frac{3}{4}$ inch, inclusive; 32.5 per cent. for a thickness above $\frac{3}{4}$ inch.

Steel plates must also be subjected to a quenching and bending test. The test piece for this must be at least 12 inches long and from 1 to $3\frac{1}{2}$ inches wide; it must be heated to a cherry red (as seen in a dark place) and then plunged into water at a temperature of about 82° F. The sample thus prepared must stand bending, without cracks or flaws, to a curve the inner radius of which is not greater than $1\frac{1}{2}$ times the thickness, and the ends must be parallel after bending.

Steel used in the construction of furnaces must have a tensile strength of not less than 58,000 nor more than 67,000

pounds per square inch, with a minimum elongation of 20 per cent. in a length of 8 inches.

6. The American rules provide that a sample taken from an iron plate must show, when tested, a tensile strength not lower than 45,000 nor higher than 60,000 pounds per square inch. The elongation in a length of 8 inches must be at least 15 per cent. The reduction in area must be as follows: For samples showing 45,000 pounds tensile strength, 15 per cent., and for each additional 1,000 pounds 1 per cent. more. Samples testing over 55,000 up to 60,000 pounds tensile strength need not show a greater reduction in area than 25 per cent.

All iron plate must be subjected to a bending test, using a test piece at least 12 inches long and from 1 to 3½ inches wide. This sample must stand bending cold, without cracks or flaws, to an angle of 90° to a curve the inner radius of which is not less than 1½ times the thickness.

7. The Board of Trade rules specify that steel must be of a general quality that has been found suitable for marine boilers. Each plate must be stamped, giving its tensile strength, and also its elongation. The elongation should be about 25 per cent. in a length of 10 inches, and must not be less than 18 per cent.; when the plate has been annealed, the elongation must not be less than 20 per cent. Plates not exposed to flame must pass a bending test; plates exposed to flame must pass a bending and quenching test, quenching in water at about 80° F. and bending cold until the sides are parallel and at a distance from each other of not more than three times the thickness. The test strip must be about 2 inches wide and 10 inches long. The tensile strength, for plates not exposed to flame, must not be less than 27 gross tons nor more than 32 gross tons per square inch. The tensile strength of furnace, flanging, and combustion box plates must range between 26 gross tons and 30 gross tons per square inch. In calculations, the Board of Trade surveyor is enjoined by the rules to take the tensile strength of approved steel plates as 27 gross tons per square inch unless all the

plates for a given boiler have been tested in his presence, in which case he may use the actual tensile strength found. The Canadian rules for steel plates are the same as the Board of Trade rules on the whole; they differ in Part II, where it is stated that in calculations made under the rules given in that Part the tensile strength of best quality steel plate is to be taken as 60,000 pounds per square inch.

8. Both the Board of Trade and Part I of the Canadian rules specify that the tensile strength of iron plates is to be taken as 47,000 pounds per square inch with the grain of the iron and as 40,000 pounds across the grain. Part II of the Canadian rules specifies that the tensile strength of iron plates made of the best material is to be taken as 48,000 pounds per square inch with the grain and 42,000 pounds per square inch across the grain. The Board of Trade rules further provide that in calculating the working pressure the actual tensile strength of tested iron plates may be used with a factor of safety of 4.5 in case the elongation in a length of 10 inches is not less than 14 per cent. with the grain and 8 per cent. across the grain, if the surveyor is satisfied as to the quality of the plates.

When iron plates are used in superheaters, the Board of Trade and Canadian rules specify that their tensile strength in calculations is to be taken as 30,000 pounds per square inch, unless the flame impinges at, or nearly at, right angles to the plate, when 22,400 pounds per square inch is to be taken as the tensile strength. The use of steel plates in superheaters is discouraged in all cases by the Board of Trade and the Canadian rules.

9. Rivets.—The Board of Trade rules prescribe that the tensile strength of the bars from which rivets are made must not be less than 26 gross tons nor more than 30 gross tons per square inch, and that the elongation must not be less than 25 per cent. in a length of 10 inches. The tensile strength of the finished rivets must not be less than 27 gross tons nor more than 32 gross tons per square inch, and they must show a reduction of area of at least 60 per cent. The

Canadian rules specify the same tensile strength and elongation for rivet bars and finished rivets as the Board of Trade rules; the American rules do not prescribe any test for rivets.

10. Stays.—The American rules specify that steel bars to be used for stays or braces must pass a cold bending test; the sample is to be bent cold to a curve the inner radius of which is equal to one and one-half times the diameter or thickness of the bar, and must stand this bending without showing flaws or cracks.

The Board of Trade and Canadian rules specify that the tensile strength of steel stay-bars must not be less than 27 gross tons nor more than 32 gross tons per square inch; the elongation should be about 25 per cent. and must not be less than 20 per cent., in a length of 10 inches. The use of welded steel stays is forbidden by the American, Board of Trade, and Canadian rules.

11. Stay-Tubes.—The tensile strength of stay-tubes, according to the Board of Trade and Canadian rules, must not be less than 26 gross tons nor more than 30 gross tons; the material should show an elongation of about 25 per cent. in 10 inches and must never be less than 20 per cent. The rules demand a minimum net thickness of $\frac{1}{4}$ inch.

12. Boiler Tubes.—The American rules provide that lap-welded boiler tubes may be made of charcoal iron, or of mild steel made by any process, and must be free from defective welds, cracks, blisters, scale pits, and sand marks. All tubes up to and including 30 inches in diameter must stand the following test: A test piece 2 inches in length cut from a tube must stand being flattened by hammering until the sides are brought parallel, with the curve on the inside at the ends not greater than three times the thickness of the metal, without showing cracks or flaws, with bend at one side being in the weld. Tubes 4 inches in diameter and under must pass a flanging test in addition, consisting of flanging the tube at right angles to the body and to a width of $\frac{3}{8}$ inch. These tests are made on cold tubes. All lap-welded boiler tubes must be tested hydrostatically to an internal pressure

of 500 pounds per square inch, and under this pressure show no signs of weakness or defects.

13. Seamless steel boiler tubes, under the American rules, must be made by the open-hearth process, must be free from all surface defects, and must have been annealed if cold-drawn.

A test piece 3 inches long cut from a tube must stand being flattened by hammering until the sides are brought parallel, with a curve on the inside at the ends not greater than three times the thickness of the metal, without showing cracks or flaws. In addition, the tubes must stand flanging all around the end to a width of $\frac{3}{8}$ inch beyond the outside body of the tube.

Both of these tests must be made on cold tubing. Each seamless steel boiler tube must be subjected to an internal hydrostatic pressure of 1,000 pounds per square inch without showing signs of weakness or defects.

14. The Board of Trade rules state: "Steel tubes made by the Mannesman process need not be objected to for use in boilers, provided the material and the tests comply in all respects with the Board's usual requirements." The Rules do not specify what these requirements are.

15. **Welded Steam and Water Pipes.**—The American rules state that welded pipe may be made of wrought iron or mild steel, and must be smooth, straight, and free from defects. The use of threaded pipe of standard thickness is discouraged and absolutely prohibited if the pipe is above 5 inches, nominal diameter.

Welded pipe up to and including $3\frac{1}{2}$ inches, nominal diameter, must be tested by an internal hydrostatic pressure to 600 pounds per square inch.

Welded pipe from 4 inches to 30 inches, nominal diameter, if made from iron must have a minimum tensile strength of 44,000 pounds per square inch and a minimum elongation of 12 per cent. in 8 inches. Welded steel pipe must have a tensile strength of not less than 50,000 pounds per square inch, and a minimum elongation of 20 per cent. All pipe

must be tested to at least 500 pounds per square inch by internal hydrostatic pressure. In addition to these requirements, a test piece 2 inches long must stand being flattened by hammering until the sides are brought parallel, with the curve on the inside at the ends not greater than 3 times the thickness of the metal, without showing cracks or flaws, with the bend at one side being in the weld.

16. Seamless Steel Steam and Water Pipes.—The American rules provide that the material for seamless steel pipe must be made by the open-hearth process, and that the pipe must be smooth and straight, and inside and outside be free from all surface defects that would materially weaken it or form starting points for corrosion. The tensile strength must not be less than 48,000 pounds per square inch, and the elongation not less than 12 per cent. in 8 inches. A test piece 2 inches long must stand, without showing cracks or flaws, being flattened by hammering until the sides are brought parallel, with the curve on the inside at the ends not greater than 3 times the thickness of the metal.

SPECIFICATIONS AND TESTS FOR CASTINGS

17. Cast Iron.—Regarding the use of cast iron, the American rules state: "No cast iron subject to pressure shall be allowed to be used in boilers or the pipes connected thereto, except as described as follows: Cast iron may be used in the construction of manhole and handhole plates, valves and cocks, water columns, flanges, saddles, ells, tees, crosses, or manifolds when such flanges, saddles, ells, tees, crosses, valves and cocks, or manifolds are bolted or riveted directly to the boiler and the valves or cocks; also casings of slip joints in pipes; provided, however, that the material shall be of the best grade and of suitable thickness and uniform section for the pressure allowed on boilers."

18. The Board of Trade rules state: "In all boilers in which the surveyors find that cast iron is employed in such a manner as to be subjected to the pressure of steam and

water, they should report the circumstances to the Board of Trade. Cast-iron stand pipes or cocks intended for the passage through them of hot brine should not be passed. Surveyors should also discourage the use of cast-iron chucks and saddles for boilers."

19. The Canadian rules state: "Cast iron must not be used for stays, and inspectors should also discourage the use of cast iron for chucks and saddles for boilers." The same rules also prohibit the use of cast iron for stays, pipes, or elbows in water-tube boilers.

20. Steel Castings.—Neither the Board of Trade nor the Canadian rules contain any special clause in regard to the use of steel castings; the American rules state: "Flowed steel castings shall possess a tensile strength of not less than 62,000 pounds, an elastic limit of not less than 30,000 pounds to the square inch, reduction of area of not less than 35 per cent., elongation of not less than 25 per cent., and contain not more than .04 per cent. of phosphorus, and not more than .03 per cent. of sulphur. Each of such castings shall be distinctly marked with name of manufacturer. Manufacturers shall furnish report of test to supervising inspector of district where castings are to be used. All steel castings shall be thoroughly annealed. Castings of steel possessing the foregoing characteristics may be used for the necks or nozzles connecting the steam drum, or dome, and the boiler, and for the fittings of boilers, and fittings of steam, feed, and water pipes: Provided, that nozzles made of cast steel shall not be used in connecting shells of externally fired boilers to mud-drums, when said nozzles are exposed to the direct action of the flame."

21. Malleable-Iron Castings.—The American rules state: "The use of malleable-iron or cast-steel manifolds, tees, return bends, or elbows in the construction of pipe generators shall be allowed."

CYLINDRICAL SHELLS

STRESSES ON CYLINDERS

22. If the cylindrical shell shown in Fig. 1 is subjected to an internal pressure, there will be two forces tending to rupture it. One force, indicated by the arrows A, A, acting in the direction of the length, tends to tear the shell in a transverse plane, as $B B_1$. The other force, indicated by the arrows C, C, acting perpendicular to the axis, tends to rupture

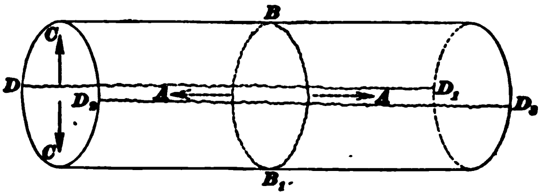


FIG. 1

the boiler in a longitudinal plane passing through the axis, as $D D_1$, $D_2 D_3$. These two forces are opposed by the tenacity of the material of which the shell is composed.

The magnitude of the force tending to rupture the shell in a transverse plane is equal to the area of the head in square inches times the steam pressure per square inch. As this force is resisted by the tenacity of the material, the magnitude of the tenacity being measured by the sectional area, the stress per square inch of section of the material is

$$\frac{\text{area of the head} \times \text{pressure}}{\text{area of section}}$$

The magnitude of the force tending to rupture the shell in Fig. 1 in a longitudinal plane is equal to the internal diameter times the length times the pressure. This force is resisted by the combined sectional area of the material of the two sides of the shell. Hence, the stress per square inch of section equals

$$\frac{\text{the internal diameter} \times \text{the length} \times \text{the pressure}}{\text{combined sectional area}}$$

23. Consider a plain cylindrical shell constructed of any convenient material. Let the inside diameter be 36 inches, the length 120 inches, the thickness of the shell ¼ inch, and the internal pressure to which it is subjected 100 pounds per square inch. The pressure on the head and, consequently, the magnitude of the force acting in the direction of the length is

$$36^2 \times .7854 \times 100 = 101{,}787.8 \text{ pounds}$$

This force is resisted by the tenacity of

$$36.5^2 \times .7854 - 36^2 \times .7854 = 28.471 \text{ square inches of material}$$

Hence, the stress per square inch of section is 101,787.8 $\div$ 28.471 = 3,575.14 pounds. The magnitude of the force acting perpendicular to the axis equals $36 \times 120 \times 100$ = 432,000 pounds. The area of the material resisting this force equals $120 \times .25 \times 2 = 60$ square inches; hence, the unit stress is 432,000 $\div$ 60 = 7,200 pounds per square inch. This shows that there is 7,200 $\div$ 3,575.14 = 2.01, say about twice as much resistance to transverse rupture as there is to rupture in a longitudinal plane. Hence, it follows that if the material is proportioned to withstand the force perpendicular to the axis, it will possess ample strength in the other direction. For convenience in calculation, the length of the shell is taken as 1 inch. If a boiler is constructed of plates varying in thickness and tensile strength, the least thickness and the lowest tensile strength must be used in calculating the strength of the boiler.

BURSTING, WORKING, AND TEST PRESSURES

24. Fundamental Rules.—In a seamless cylinder that is on the point of bursting, the resistance of the material to rupture must be equal to the force tending to cause rupture. Hence, such a cylinder is on the point of bursting if the product of the diameter and pressure equals the product of twice the thickness of the cylinder and the ultimate tensile strength of the material of which it is composed. From this, it follows that the bursting pressure equals

$$\frac{\text{twice the thickness} \times \text{the ultimate tensile strength}}{\text{diameter}}$$

This may be simplified by using the radius of the cylinder instead of the diameter. Then, as the radius is one-half the diameter, the bursting pressure will be

$$\frac{\text{the thickness} \times \text{the ultimate tensile strength}}{\text{radius}}$$

Rule.—*To find the bursting pressure of a seamless cylinder, in pounds per square inch, divide the product of its thickness, in inches, and the ultimate tensile strength, in pounds per square inch, by the internal radius, in inches.*

Or, $$P_b = \frac{TS}{R}$$

in which T = thickness of shell of cylinder, in inches;
R = internal radius, in inches;
S = ultimate tensile strength of material, in pounds per square inch;
P_b = bursting pressure, in pounds per square inch.

EXAMPLE.—A cylinder 48 inches in internal diameter and .375 inch thick is made of wrought iron having a tensile strength of 50,000 pounds per square inch; what is its bursting pressure?

SOLUTION.—The internal radius is 48 ÷ 2 = 24 in. Applying the rule,

$$P_b = \frac{.375 \times 50{,}000}{24} = 781.25 \text{ lb. per sq. in. Ans.}$$

25. When a cylinder has a longitudinal seam or joint, its bursting strength is diminished. It is usual to express the efficiency of a seam or joint in per cent. of the solid plate.

Rule.—*To find the bursting strength, in pounds per square inch, of a cylinder having a longitudinal seam or joint, divide the product of the thickness, in inches, the tensile strength of the material, in pounds per square inch, and the efficiency of the seam or joint expressed decimally, by the internal radius, in inches.*

Or, $$P_b = \frac{TSE}{R}$$

in which E is the efficiency of the joint, expressed decimally, and the other letters have the same meaning as in the formula given in Art. **24.**

EXAMPLE.—A boiler shell 60 inches in diameter is constructed of material having a tensile strength of 60,000 pounds per square inch and .5 inch thick. It has a single-riveted longitudinal joint having an efficiency of 56 per cent.; at what pressure will the shell burst?

SOLUTION.—The internal radius is 60 ÷ 2 = 30 inches. Applying the rule,

$$P_b = \frac{.5 \times 60{,}000 \times .56}{30} = 560 \text{ lb. per sq. in.} \quad \text{Ans.}$$

26. The fundamental rules given in Arts. **24** and **25** are used in the Board of Trade, Canadian, and American rules as a basis for determining the safe working pressure on boiler shells and pipes subjected to internal pressure, either by introducing a factor of safety or a coefficient combining a factor of safety with the efficiency of the longitudinal seam or joint. A **factor of safety**, when referring to boilers, may be defined as the ratio between the safe working and the bursting pressure.

27. American Rule for Working Pressure on Boiler Shell.—The American rules provide that the working pressure allowable on a boiler shell shall be ascertained as follows:

Rule.—*Multiply one-sixth of the lowest tensile strength found stamped on any plate in the cylindrical shell by the thickness, in inches, of the thinnest plate in the shell and divide by the radius, in inches; the quotient will be the pressure allowable per square inch for single riveting, to which add 20 per cent. for double riveting, when all the rivet holes in the shell of such a boiler have been "fairly drilled" and no part of such holes has been punched.*

Or, for single-riveted longitudinal joints,

$$P_w = \frac{Tt}{6R}$$

and for double-riveted longitudinal joints,

$$P_w = \frac{1.2\,Tt}{6R} = \frac{Tt}{5R}$$

in which P_w = working pressure, in pounds per square inch;
T = tensile strength, in pounds per square inch;
t = thickness, in inches;
R = radius, in inches.

The factors 5 and 6 appearing in the formulas are not factors of safety, but coefficients; making allowance for the weakening effect of the seams, these coefficients usually represent factors of safety of 3.5 and 3.4, approximately.

EXAMPLE 1.—A boiler 48 inches in diameter, with single-riveted seams, is constructed of material $\frac{3}{8}$ inch thick and having a tensile strength of 50,000 pounds per square inch; what working pressure will be allowed on the boiler shell?

SOLUTION.—Applying the rule,

$$P_w = \frac{\frac{3}{8} \times 50{,}000}{6 \times \frac{48}{2}} = 130.2 \text{ lb. per sq. in. Ans.}$$

EXAMPLE 2.—A Scotch boiler is 14 feet 2 inches in diameter; the shell plates are of steel having a tensile strength of 62,000 pounds per square inch and are 1 inch thick. The longitudinal seams are triple riveted, with inner and outer butt straps, and in the American rules are considered as equivalent, for the purpose of calculation, to double-riveted seams. What working pressure will be allowed on the shell?

SOLUTION.— 14 ft. 2 in. = 170 in. The radius is 170 ÷ 2 = 85 in. Applying the rule,

$$P_w = \frac{62{,}000 \times 1}{5 \times 85} = 145.9 \text{ lb. per sq. in., nearly. Ans.}$$

28. Board of Trade and Canadian Rules for Working Pressure on Boiler Shell.—Both parts of the Canadian rules, and also the Board of Trade rules provide that the allowable working pressure on a boiler shell is to be ascertained as follows:

Rule.—*Multiply the tensile strength of the material, in pounds per square inch, by the least efficiency of the longitudinal joint, in per cent. expressed decimally, by 2, and by the plate thickness, in inches. Divide the product by the product of the factor of safety and inside diameter of the boiler, in inches. The quotient will be the allowable pressure, in pounds per square inch, on the boiler shell.*

Or,

$$B = \frac{S\% 2T}{FD}$$

in which B = working pressure, in pounds per square inch;
S = tensile strength, in pounds per square inch;

% = least efficiency of longitudinal joint, in per cent., expressed decimally;
T = thickness of plate, in inches;
D = diameter of boiler, in inches;
F = factor of safety.

EXAMPLE.—What working pressure is allowable on a boiler shell 14 feet 2 inches in diameter, $1\frac{1}{4}$ inches thick, constructed of steel plates having a tensile strength of 60,000 pounds per square inch, and a joint efficiency of 80 per cent., using a factor of safety of 4.8?

SOLUTION.— 14 ft. 2 in. = 170 in. Applying the rule,

$$B = \frac{60{,}000 \times .80 \times 2 \times 1.25}{4.8 \times 170} = 147 \text{ lb. per sq. in., nearly. Ans.}$$

29. It is to be noted in regard to the tensile strength of the material that the surveyor or inspector is enjoined by the Rules to use the actual tensile strength of iron or steel plate only in case all the plates have been actually tested; in all other cases, both in the Board of Trade rules and those in Part I of the Canadian rules, iron plates are to be assumed to have a tensile strength of 47,000 pounds per square inch (48,000 pounds in Part II, Canadian rules) with the grain, and 40,000 pounds per square inch (42,000 pounds in Part II, Canadian rules) across the grain. Steel plates under the same conditions are assumed to have a tensile strength of 27 gross tons per square inch; their actual tensile strength may be used only in case the surveyor or inspector has personally witnessed the testing of all the plates.

30. The Board of Trade rules state: "When the cylindrical shells of boilers are made of best material with all the rivet holes drilled in place and all the seams fitted with double butt straps each of at least five-eighths the thickness of the plates they cover, and all the seams at least double riveted with rivets having an allowance of not more than 75 per cent. over the single shear, and provided that the boilers have been open to inspection during the whole period of inspection, then 5 may be taken as the factor of safety."

"If, however, the iron be tested and the elongation measured in a length of 10 inches is not less than 14 per cent. with, and 8 per cent. across, the grain, and the surveyors are

TABLE I

ADDITIONS TO FACTOR OF SAFETY ON BOILER SHELLS

Identification Letter	Add	Condition
A†	.15	To be added when all the holes are fair and good in the longitudinal seams, but drilled out of place after bending.
B†	.3	To be added when all the holes are fair and good in the longitudinal seams, but drilled before bending.
C	.3	To be added when all the holes are fair and good in the longitudinal seams, but punched after bending.
D	.5	To be added when all the holes are fair and good in the longitudinal seams, but punched before bending.
*E**	.75	To be added when all the holes are not fair and good in the longitudinal seams.
F	.1	To be added if the holes are all fair and good in the circumferential seams, but drilled out of place after bending.
G†	.15	To be added if the holes are fair and good in the circumferential seams, but drilled before bending.
H	.15	To be added if the holes are fair and good in the circumferential seams, but punched after bending.
I†	.2	To be added if the holes are fair and good in the circumferential seams, but punched before bending.
*J**	.2	To be added if the holes are not fair and good in the circumferential seams.
K	.2	To be added if double butt straps are not fitted to the longitudinal seams and the said seams are lapped and double riveted.
L	.1	To be added if double butt straps are not fitted to the longitudinal seams, and the said seams are lapped and treble riveted.
M	.3	To be added if only single butt straps are fitted to the longitudinal seams, and the said seams are double riveted.

TABLE I—*Continued*

Identification Letter	Add	Condition
N	.15	To be added if only single butt straps are fitted to the longitudinal seams, and the said seams are treble riveted.
O	1.0	To be added when any description of joint in the longitudinal seams is single riveted.
*P***	.1	To be added if the circumferential seams are fitted with single butt straps and are double riveted.
*Q***	.2	To be added if the circumferential seams are fitted with single butt straps and are single riveted.
*R***	.1	To be added if the circumferential seams are fitted with double butt straps and are single riveted.
*S***‡	.1	To be added if the circumferential seams are lapped and double riveted.
T	.2	To be added if the circumferential seams are lapped and single riveted.
U	.25	To be added when the circumferential seams are lapped and the strakes of plates are not entirely under or over.
V‡	.3	To be added when the boiler is of such a length as to fire from both ends, or is of unusual length, as in the case of flue boilers, and the circumferential seams fitted as described opposite *P*, *R*, and *S*; but when the circumferential seams are as described opposite *Q* and *T*, .4 should be added.
*W**	.4	To be added if the longitudinal seams are not properly crossed.
*X**	.4	To be added when the iron is in any way doubtful and the surveyor (inspector) is not satisfied that it is of the best quality.
Y‡‡	1.65	To be added if the boiler is not open to inspection during the whole period of its construction.
Y‡‡	1.0	Part II, Canadian rules: To be added if the boiler is not open to inspection during the whole period of its construction.

NOTES IN REGARD TO TABLE I

†Board of Trade, but not Canadian, rules: "When the holes are to be rimered or bored out in place the case should be submitted to the Board as to the reduction or omission of *A*, *B*, *G*, and *I*."

*Board of Trade and both parts of Canadian rules: "The factor may be increased still further if the workmanship or material is such as in the surveyor's (inspector's) judgment renders such increase necessary."

**Both parts of Canadian, but not Board of Trade, rules: "Shall not apply to the end or circumferential seams if such seams are sufficiently stayed by through bolts; nor to the seams between the square and round part of the shell, in cylindrical boilers with square furnaces, when such seams are double riveted."

‡Board of Trade, but not Canadian, rules: "When the middle circumferential seams are double strapped and double riveted or lapped and treble riveted, and the calculated strength not less than 65 per cent. of the solid plate, *S* and *V* may be omitted. The end circumferential seams in such cases should be at least double riveted."

‡‡Both parts of Canadian, and Board of Trade, rules: "When surveying (inspecting) boilers that have not been open to inspection during construction, the case should be submitted to the Board (Chairman, Board of Steamboat Inspection, in Canada) as to the factors to be used."

otherwise satisfied as to the quality of the plates and rivets, 4.5 may be used as the factor of safety instead of 5, in which case the minimum actual tensile strength of the plates should be used in calculating the working pressure."

The Canadian rules prevent the use of untested boiler plates; Part I establishes 4.5 as the factor of safety and Part II names 4.

Both parts of the Canadian rules and also the Board of Trade rules state: "When the above conditions are not complied with, the additions in the following scale should be made to the factor of safety, according to the circumstances of each case."

The object of these additions to the factor of safety is to promote good workmanship and design.

EXAMPLE.—What pressure would a Board of Trade surveyor allow on the shell of a Scotch boiler having a diameter of 12 feet, a thickness of 1 inch, and entirely made of inspected iron plate having a tensile strength of 50,000 pounds per square inch? The longitudinal seams are lapped and double riveted, with iron rivets $1\frac{5}{16}$ inch diameter and $4\frac{1}{16}$ inch pitch, have an efficiency of 66.6 per cent., and the rivet holes have been punched fair and good after bending. The circumferential seams are lapped, fair and well punched after bending, and

are single riveted. The boiler has been open to inspection and the surveyor is satisfied with the quality of the material.

Solution.—According to Art. **30**, the factor of safety, the plate having been tested, is 4.5. This is to be increased, for lap joints and double riveting, by .2 (see K, Table I); and by .3 for punching in the longitudinal seams after bending (see C, Table I); and by .2 for circumferential seams lapped and single riveted (see T, Table I); and by .15 for punching the holes in the circumferential seams (see H, Table I). This makes the factor of safety

$$4.5 + .2 + .3 + .2 + .15 = 5.35$$

Applying the rule in Art. **28**,

$$B = \frac{50{,}000 \times .666 \times 2 \times 1}{5.35 \times 12 \times 12} = 86.45 \text{ lb. per sq. in. Ans.}$$

31. In regard to steel boiler shells, the Board of Trade rules state: "When the minimum tensile strength of the shell plates is S tons and full allowance is wished, the rivet section, if iron, in the longitudinal seams of cylindrical shells should, when those seams are lapped, be at least $\frac{S}{17.5}$ the net plate section, and if steel rivets are used their section should be at least $\frac{S}{23}$ of the net section of the plate if the tensile strength of the rivets is not less than 27 tons (gross) and not more than 32 tons (gross) per square inch. In calculating the working pressure, the percentage strength of the rivets may be found in the usual way by the Board's rules, but in dealing with iron rivets the percentages found should be divided by $\frac{S}{17.5}$, and in the case of steel rivets by $\frac{S}{23}$, the results being the percentages required. If the percentage strength of the rivets is found by calculation to be less than the calculated percentage strength of the plate, the working pressure should be calculated by both percentages. When using the percentage strength of the plate 4.5 plus the additions suitable for the method of construction as by the Board's rules for iron boilers, may be used as the nominal factor of safety, but when using the percentage strength of the rivets 4.5 may be used as the factor of safety. The less of the two pressures so found is the working pressure to be allowed for the cylindrical portion of the shell."

32. The Canadian rules state in regard to steel boiler shells: "The rivet section, if of iron, in the longitudinal seams of cylindrical shells, where lapped and at least double riveted, should not be less than $\frac{13}{8}$ times the net plate section; but if steel rivets are used, their section should be at least $\frac{23}{28}$ of the net section of the plate if the tensile strength of the rivet is not less than 27 tons gross and not more than 32 tons gross per square inch. Therefore, in calculating the working pressure, the percentage strength of the rivets may be found in the usual way by the rules, but in the case of iron rivets the percentages found should be divided by $\frac{13}{8}$, and in the case of steel rivets by $\frac{23}{28}$, the result being the percentages required. If the percentage strength of the rivets by calculations is less than the calculated percentage strength of the plate, calculate the working pressure by both percentages. When using the percentage strength of the plate, 4.25 plus the additions suitable for the method of construction as by the rules for iron boilers may be used as the nominal factor of safety, but when using the percentage strength of the rivets, 4.25 may be used as the factor of safety. The less of the two pressures so found is the working pressure to be allowed for the cylindrical portion of the shell, or otherwise in accordance with the formulas in the appendix."

The formulas referred to are given under the heading Riveted Joints.

33. The working pressure allowable on the shell of cylindrical superheaters, under the Board of Trade and Canadian rules, is to be calculated in the same manner as for a boiler shell, except that, as previously stated, the tensile strength of the iron must be taken as 30,000 pounds per square inch, unless the heat or flame impinges at, or nearly at, right angles to the plate, when 22,400 pounds per square inch is to be used as the tensile strength.

34. The working pressure of the shells of steam drums and mud-drums, except those of water-tube boilers, is calculated by the same rules governing that of the boiler shell.

35. Working Pressure Allowable on Drums of Water-Tube Boilers.—The Board of Trade rules do not contain any specifications in regard to water-tube boilers; the Canadian rules specify that the working pressure on water-tube boiler drums exposed to the fire is to be found by the rule given in Art. **28**, taking the strength of the plate as 30,000 pounds per square inch and the factor of safety as 5, making additions thereto, if necessary, as specified in Art. **30** and Table I, and calculating the efficiency of the joint and also the percentage of plate left by the line of holes where the water tubes are attached, using the lowest percentage found.

When the drum of a water-tube boiler is not exposed to flame, under Canadian rules, the calculation is made in the same manner as for a boiler shell, using 5 as the factor of safety, with additions such as the conditions specified in Art. **30** require, and making allowance for the weakening due to the holes receiving the water tubes.

36. The American rules state that the working pressure allowable on the shell of a drum forming part of a water-tube or coil boiler, when such shell has a row or rows of pipes or tubes inserted therein, shall be determined as follows:

Rule.—*From the distance, in inches, between the tube or pipe centers, in a line from head to head, subtract the diameter of the tube hole, in inches. Multiply the remainder by the thickness of the plate, in inches, and by one-sixth of its tensile strength. Divide the product by the product of the distance, in inches, between the tube or pipe centers in a line from head to head, and the radius of the shell, in inches. The quotient will be the allowable working pressure, in pounds per square inch, on the shell of a water-tube boiler drum.*

Or,
$$P_w = \frac{(D-d)\,T\,S}{D\,R}$$

in which P_w = allowable working pressure, in pounds per square inch;

D = distance between tube or pipe centers in a line from head to head, in inches;

d = diameter of hole, in inches;

T = thickness of plate, in inches;
S = one-sixth of tensile strength;
R = radius of shell, in inches.

EXAMPLE.—The drum of a water-tube boiler has a diameter of 24 inches, is $\frac{1}{2}$ inch thick, and is constructed of steel having a tensile strength of 60,000 pounds per square inch. The water tubes are $1\frac{1}{4}$ inches outside diameter and spaced 2 inches center to center. What working pressure is allowable on the shell under American rules?

SOLUTION.— $S = 60,000 \div 6 = 10,000$. $R = 24 \div 2 = 12$. Applying the rule,

$$P_w = \frac{(2 - 1\frac{1}{4}) \times .5 \times 10,000}{2 \times 12} = 156.25 \text{ lb. per sq. in. Ans.}$$

37. Hydrostatic Test Pressures.—The American rules provide that fire-tube boilers must be subjected to a hydrostatic test of $1\frac{1}{2}$ times the working pressure, and water-tube boilers must be tested to twice the working pressure.

The Canadian rules specify the same tests as the American rules.

The Board of Trade rules state that all boilers shall be subjected to a hydrostatic test of twice the working pressure, and that no hydrostatic test shall be considered good in which the boiler has not borne satisfactorily the intended test pressure for at least 10 consecutive minutes.

EXAMPLES FOR PRACTICE

1. What is the bursting pressure of a seamless steel tube 10 inches in diameter and $\frac{1}{4}$ inch thick if the tensile strength is 65,000 pounds per square inch? Ans. 3,250 lb. per sq. in.

2. A boiler shell 48 inches in diameter and $\frac{3}{8}$ inch thick has a tensile strength of 55,000 pounds per square inch. The efficiency of the joint being 70 per cent., at what pressure would the shell burst? Ans. 601.56 lb. per sq. in.

3. What working pressure, under American rules, will be allowed on a 48-inch boiler shell $\frac{3}{8}$ inch thick, double riveted, and having a tensile strength of 50,000 pounds per square inch? Ans. 156.25 lb. per sq. in.

4. Under Board of Trade rules, what working pressure will be allowed on the boiler shell in example 3 if the factor of safety is 5 and the efficiency of the joint 70 per cent.? Ans. 109.38 lb. per sq. in.

5. A water-tube boiler has a drum 20 inches in diameter and $\frac{7}{16}$ inch thick, the plate having a tensile strength of 60,000 pounds per square inch. The water tubes have an outside diameter of 2 inches and are spaced 4 inches center to center. What working pressure will be allowed under American rules? Ans. 218.75 lb. per sq. in.

6. A water-tube boiler is to be worked at 200 pounds per square inch; what must the hydrostatic test pressure be under American rules? Ans. 400 lb. per sq. in.

RIVETED JOINTS

GENERAL REGULATIONS

38. The American rules state: "All boilers for marine purposes shall be required to have the rivet holes in the shells, heads, and flanges of same, steam and mud-drums, and holes for stay-bolts and tubes fairly drilled, and no part of such holes shall be punched."

The Board of Trade and Canadian rules permit the punching of holes, penalizing this method, however, by requiring a higher factor of safety to be used. See Table I.

According to the Board of Trade and Canadian rules, rivets in double shear are considered as having 1.75 times the shearing strength of rivets in single shear.

Rivets, before riveting, are usually about $\frac{1}{16}$ inch smaller in diameter than the rivet hole, to permit easy insertion; when riveted they fill the hole. In all rules and formulas involving the use of either the diameter or area of rivets, the diameter or area after riveting is referred to.

The Board of Trade and Canadian rules specify that when plates, including butt straps, have been drilled in place, the plates must be taken apart after drilling, the burr taken off, and the holes slightly countersunk from the outside.

BUTT STRAPS

39. Under American rules, single butt straps must not be thinner than the plate; double butt straps must be at least five-eighths the plate thickness. The Board of Trade and Canadian rules demand single butt straps to be at least one

and one-eighth the plate thickness; double butt straps must be at least five-eighths the plate thickness. The same rules specify that butt straps must be cut from plates, be of as good quality as the shell plates, and those for the longitudinal seams should be cut across the fiber. Both the Board of Trade and Canadian rules specify that when the joint has double the number of rivets in the inner than in the outer row, that is, when every alternate rivet in the outer row has been omitted, the least thickness of the butt straps must be found as follows:

Rule I.—*To find the least thickness of a single butt strap with a joint having every alternate rivet omitted in the outer row, multiply 9 by the plate thickness, in inches, and by the difference between the pitch and the rivet diameter. Divide the product by 8 times the difference between the greatest pitch and twice the rivet diameter.*

Or, $$T_1 = \frac{9\,T(p-d)}{8\,(p-2d)} \qquad (1)$$

in which T_1 = thickness of butt strap, in inches;
T = thickness of plate, in inches;
p = greatest pitch of rivets, in inches;
d = diameter of rivet, in inches.

Rule II.—*To find the least thickness of double butt straps with a joint having every alternate rivet omitted in the outer row, multiply 5 by the plate thickness, in inches, and by the difference between the pitch and the rivet diameter. Divide the product by 8 times the difference between the pitch and twice the rivet diameter.*

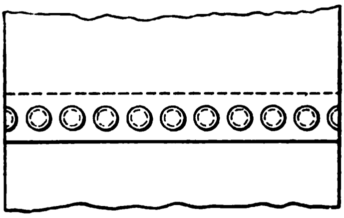
Or, $$T_1 = \frac{5\,T(p-d)}{8\,(p-2d)} \qquad (2)$$

in which the letters have the same meaning as in formula 1.

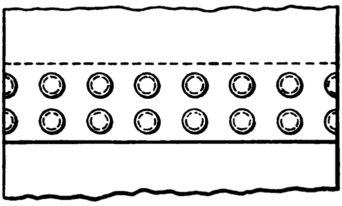
EXAMPLE 1.—A single-butt-strap joint is triple riveted, the pitch of the rivets in the outer row being 9 inches and in the inner rows $4\frac{1}{2}$ inches; the plate is $1\frac{1}{4}$ inches thick and the rivets are $1\frac{1}{2}$ inches in diameter. What is the least thickness of the butt strap?

SOLUTION.—Applying rule I,

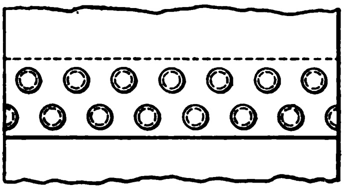
$$T_1 = \frac{9 \times 1\frac{1}{4} \times (9 - 1\frac{1}{2})}{8 \times (9 - 2 \times 1\frac{1}{2})} = 1.758 \text{ in. Ans.}$$



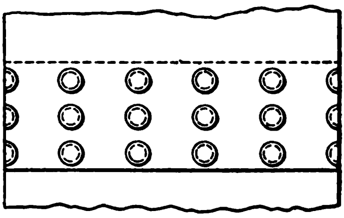
Single-Riveted Lap Joint



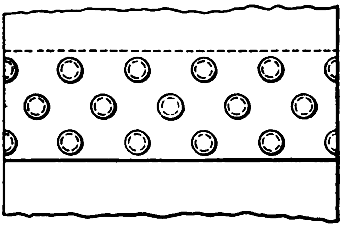
Double-Chain-Riveted Lap Joint



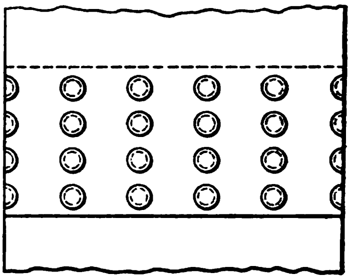
Double-Zigzag-Riveted Lap Joint



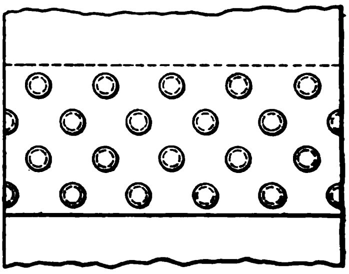
Triple-Chain-Riveted Lap Joint



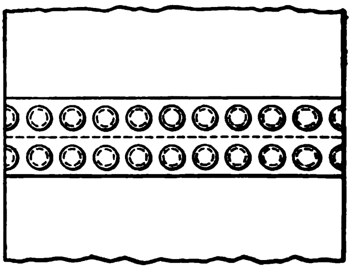
Triple-Zigzag-Riveted Lap Joint



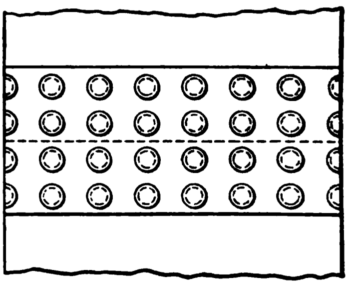
Quadruple-Chain-Riveted Lap Joint



Quadruple-Zigzag-Riveted Lap Joint



Single-Riveted Butt Joint



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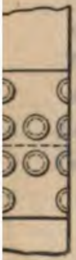

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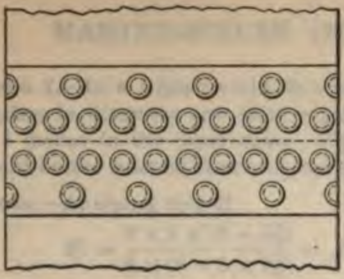
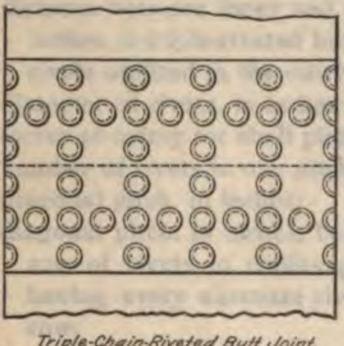


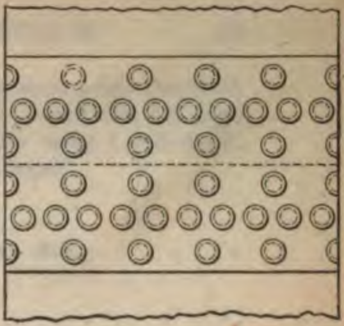
Double-Zigzag-Riveted Butt Joint.
Alternate Rivets Omitted in Outer Row.

Triple-Chain-Riveted Lap Joint.
Alternate Rivets Omitted in Outer Rows.

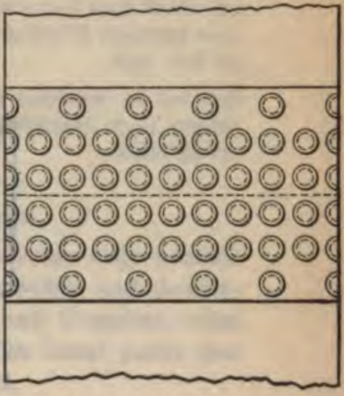
Triple-Zigzag-Riveted Lap Joint.
Alternate Rivets Omitted in Outer Rows.



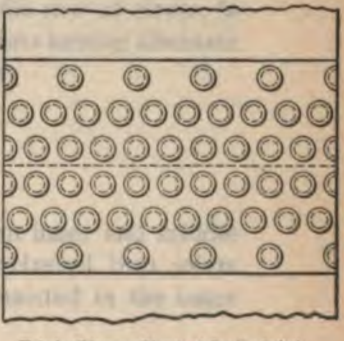
Triple-Chain-Riveted Butt Joint.
Alternate Rivets Omitted in Inner and Outer Rows.



Triple-Zigzag-Riveted Butt Joint.
Alternate Rivets Omitted in Inner and Outer Rows.



Triple-Chain-Riveted Butt Joint.
Alternate Rivets Omitted in Outer Row.



Triple-Zigzag-Riveted Butt Joint
Alternate Rivets Omitted in Outer Row

EXAMPLE 2.—In a triple-riveted double-butt-strap joint, the rivets are $1\frac{1}{4}$ inches in diameter and have a pitch of 8 inches in the outer row and 4 inches in the inner rows. The plate being 1 inch thick, what should be the least thickness of the butt straps?

SOLUTION.—Applying rule II,

$$T_1 = \frac{5 \times 1 \times (8 - 1\frac{1}{4})}{8 \times (8 - 2 \times 1\frac{1}{4})} = .767 \text{ in. Ans.}$$

EXAMPLES FOR PRACTICE

1. Under Canadian rules, in a double-butt-strap triple-riveted joint having every alternate rivet omitted in the outer row, what is the least thickness of the butt straps? The plate is $\frac{7}{8}$ inch thick; the rivets are $1\frac{1}{16}$ inches in diameter and pitched 7 inches in the outer row.
Ans. .666 in.

2. Under American rules, what is the least thickness of the butt strap in a double-butt-strap joint for a plate thickness of $1\frac{1}{4}$ inches?
Ans. $\frac{25}{32}$ in.

PROPORTIONS OF JOINTS

40. Introductory.—The American rules only contain formulas for the proportions of single-riveted and double-riveted lap joints; the Board of Trade and Canadian rules give formulas for butt joints as well. The usual joints met with in marine boilers are given in Fig. 2.

In the formulas given here, the letters have the following meaning:

V = distance between rows of rivets, in inches;
V_1 = distance between inner and middle row of rivets, in inches, in triple-riveted butt joints having alternate rivets omitted in the outer row;
d = diameter of rivets, in inches;
F = factor of safety for shell plates;
n = number of rivets in one pitch;
p_d = diagonal pitch, in inches;
P_d = diagonal pitch, in inches, between inner and middle row of rivets in triple-zigzag-riveted butt joints having every alternate rivet omitted in the outer row;

p = greatest pitch of rivets, in inches;
S_1 = tensile strength of plate, in tons of 2,240 pounds;
T = thickness of plate, in inches.

The number n of rivets in one pitch is an expression requiring an explanation. Conceive the joint to be divided by lines at right angles to itself into equal strips having a width equal to the greatest pitch of the rivets. Then, the number of rivets in one of these strips is the number referred. In case of butt joints, the number of rivets at one side of the joint is taken.

In a single-riveted lap or butt joint $n = 1$; in a double-riveted lap or butt joint, either chain or zigzag riveted, $n = 2$; in a triple-riveted lap or butt joint, either chain or zigzag riveted, $n = 3$; in a quadruple-riveted lap joint, either chain or zigzag riveted, $n = 4$; in a double-riveted butt joint, either chain or zigzag riveted, with every alternate rivet omitted in the outer row, $n = 3$; in a triple-riveted lap joint, either chain or zigzag riveted, with every alternate rivet omitted in the two outer rows, $n = 4$; in a triple-riveted butt joint, either chain or zigzag riveted, where every alternate rivet has been omitted in the inner and outer rows, $n = 4$; in a triple-riveted butt joint, either chain or zigzag riveted, where every alternate rivet has been omitted in the outer row, $n = 5$.

41. Rules for Pitch of Rivets.—Under American rules, to find the pitch of rivets in single-riveted or double-riveted lap joints, either chain or zigzag riveted, for iron plates and iron rivets, proceed as follows:

Rule.—*Square the diameter of the rivet, in inches, multiply by .7854, and also by the number of rivets in one pitch. Divide the product by the thickness of the plate, in inches, and to the quotient add the diameter of the rivet.*

Or,
$$p = \frac{d^2 \, .7854 \, n}{T} + d$$

EXAMPLE.—What is the pitch for a double-riveted lap joint having rivets $\frac{15}{16}$ inch in diameter, the plate being $\frac{5}{8}$ inch thick? Both the plate and the rivets are iron.

SOLUTION.—Applying the rule,

$$p = \frac{(\frac{15}{16})^2 \times .7854 \times 2}{\frac{5}{8}} + \frac{15}{16} = 3.146 \text{ in. Ans.}$$

42. For steel plate and steel rivets, the pitch, under the American rules, is found as follows for single-riveted or double-riveted lap joints:

Rule.—*Multiply 23 by the square of the rivet diameter, in inches, and by .7854 and by the number of rivets in one pitch. Divide the product by 28 times the thickness of the plate, in inches, and to the quotient add the rivet diameter.*

Or,
$$p = \frac{23\, d^2\, .7854\, n}{28\, T} + d$$

NOTE.—This formula reduces to the simpler form $p = \frac{.645\, d^2\, n}{T} + d$; examiners of marine engineers usually prefer candidates to work examples by the formulas printed in the Rules of their various Boards, which formulas for this reason are here given as they appear in the American, Board of Trade, and Canadian rules. Many of these formulas can be reduced mathematically to a simpler form.

EXAMPLE.—Find the pitch of steel rivets $\frac{7}{8}$ inch in diameter for a single-riveted lap joint, with steel plate $\frac{7}{16}$ inch thick.

SOLUTION.—Applying the rule,

$$p = \frac{23 \times (\frac{7}{8})^2 \times .7854 \times 1}{28 \times \frac{7}{16}} + \frac{7}{8} = 2.004 \text{ in. Ans.}$$

43. To prevent choosing a rivet diameter larger in proportion to the thickness of the plate than experience has shown to be warranted, the American rules provide that in single-riveted lap joints the pitch of the rivets must never be greater than 1.31 times the plate thickness plus $1\frac{5}{8}$ inches. For double-riveted lap joints, the pitch must not exceed 2.62 times the plate thickness plus $1\frac{5}{8}$ inches.

44. The Board of Trade and Canadian rules for finding the greatest pitch of rivets in ordinary chain or zigzag-riveted lap or butt joints for iron plates with iron rivets are alike.

Rule.—*Multiply the square of the rivet diameter by .7854 and by the number of rivets in one pitch, and by 1.75 if the rivets are in double shear. Divide the product by the plate thickness and to the quotient add the rivet diameter.*

Or, for single shear,

$$p = \frac{d^2 .7854 n}{T} + d$$

and for double shear,

$$p = \frac{d^2 .7854 n 1.75}{T} + d$$

EXAMPLE.—What is the greatest pitch of the rivets in an ordinary triple-riveted, double-butt-strap joint where the plate is 1 inch thick and the rivets are $1\frac{1}{16}$ inches in diameter, both plate and rivets being of iron?

SOLUTION.—In this case $n = 3$; see Art. **40**. The rivets are in double shear. Applying the rule,

$$p = \frac{(1\frac{1}{16})^2 \times .7854 \times 3 \times 1.75}{1} + 1\frac{1}{16} = 5.717 \text{ in.} \quad \text{Ans.}$$

45. The Board of Trade and Canadian rules for finding the greatest pitch of rivets in ordinary chain or zigzag-riveted lap or butt joints for steel plates with steel rivets are almost alike, differing slightly in the numerical value of one of the factors.

Rule.—*Multiply 23 by the square of the diameter of the rivet, in inches, and by .7854, and the number of rivets in one pitch, and by 1.75 if the rivets are in double shear, and by the factor of safety for shell plates. Divide the product by the product of 4.5 (4.25 under Canadian rules) and the tensile strength of the plate, in gross tons, and the thickness of the plate, in inches. To the quotient add the diameter of the rivet, in inches.*

Or, for single shear (Board of Trade),

$$p = \frac{23 d^2 .7854 n F}{4.5 S_1 T} + d$$

and for double shear (Board of Trade),

$$p = \frac{23 d^2 .7854 n 1.75 F}{4.5 S_1 T} + d$$

and for single shear (Canada),

$$p = \frac{23 d^2 .7854 n F}{4.25 S_1 T} + d$$

and for double shear (Canada),

$$p = \frac{23 d^2 .7854 n 1.75 F}{4.25 S_1 T} + d$$

EXAMPLE.—Under Board of Trade rules, what is the greatest pitch of rivets in a double-butt-strap joint, double chain riveted with 1-inch rivets, if the plate is $\frac{3}{4}$ inch thick, has a tensile strength of 28 gross tons, and the factor of safety corresponding to the construction is 5.3? Plate and rivets are steel.

SOLUTION.—By Art. **40**, $n = 2$. Applying the rule for double shear,

$$p = \frac{23 \times 1^2 \times .7854 \times 2 \times 1.75 \times 5.3}{4.5 \times 28 \times \frac{3}{4}} + 1 = 4.55 \text{ in., nearly. Ans.}$$

46. To prevent the choice of a diameter of rivet so large as to be badly out of proportion to the thickness of the plate, both the Board of Trade and Canadian rules fix a maximum pitch that must never be exceeded.

Rule.—*To find the maximum pitch of rivets, multiply the constant taken from Table II by the thickness of the plate, in inches, and add $1\frac{5}{8}$ inches.*

Or, $$p = CT + 1\frac{5}{8}$$
in which C = constant taken from Table II.

TABLE II

CONSTANTS FOR FINDING MAXIMUM PITCH OF RIVETS

Number of Rivets in One Pitch	Constants for Lap Joints	Constants for Double-Butt-Strap Joints
1	1.31	1.75
2	2.62	3.50
3	3.47	4.63
4	4.14	5.52
5		6.00

It will be plain that the pitch of rivets should be calculated by the proper one of the preceding rules and then checked by applying the rule given in this article.

EXAMPLE.—Referring to the example in Art. **45**, find if the pitch there calculated will be passed.

SOLUTION.—By Table II, the constant is 3.5. Applying the rule,

$$p = 3.5 \times \tfrac{3}{4} + 1\tfrac{5}{8} = 4.25 \text{ in.}$$

This shows that the pitch previously calculated is too large to pass inspection. Ans.

47. Rules for Diameter of Rivet.—The American rules present a table of proportions of single- and double-riveted lap joints, from which it appears that for single-riveted lap joints and iron plates and iron rivets the rivet diameter is equal to the plate thickness plus $\frac{3}{8}$ inch. For double-riveted lap joints and iron plates and iron rivets, the rivet diameter is equal to the plate thickness plus $\frac{5}{16}$ inch. For single-riveted lap joints and steel plates and steel rivets, the rivet diameter is equal to the plate thickness plus $\frac{7}{16}$ inch. For double-riveted lap joints and steel plates and steel rivets, the rivet diameter is equal to the plate thickness plus $\frac{3}{8}$ inch.

The Board of Trade and Canadian rules state that the rivet diameter should never be less than the thickness of the plate.

48. Rules for Distance From Center of Rivet to Edge of Plate.—The American, Board of Trade, and Canadian rules all state that the distance from the center of the rivet to the edge of the plate must never be less than $1\frac{1}{2}$ times the rivet diameter.

49. Rules for Distance Between Rows of Rivets. The American, Board of Trade, and Canadian rules specify that the distance between the rows of rivets in all ordinary chain-riveted joints (the American rules only cover double-chain-riveted lap joints) shall not be less than twice the rivet diameter, and should preferably be twice the rivet diameter plus $\frac{1}{2}$ inch.

50. For double-zigzag-riveted lap joints, the American rules specify that the distance between the rows of rivets must be that given by applying the rule in this article. The Canadian and Board of Trade rules prescribe the same rule for all ordinary zigzag-riveted lap and butt joints.

Rule.—*Multiply the sum of 11 times the pitch and 4 times the rivet diameter by the sum of the pitch and 4 times the rivet diameter. Extract the square root of the quotient and divide the root by 10.*

Or,
$$V = \frac{\sqrt{(11\,p + 4\,d)\,(p + 4\,d)}}{10}$$

EXAMPLE.—With rivets having a diameter of $\frac{3}{4}$ inch and a pitch of $2\frac{1}{2}$ inches, what should be the distance between the rows of rivets, the joint being double zigzag riveted?

SOLUTION.—Applying the rule,

$$V = \frac{\sqrt{(11 \times 2\frac{1}{2} + 4 \times \frac{3}{4}) \times (2\frac{1}{2} + 4 \times \frac{3}{4})}}{10} = 1.295 \text{ in. Ans.}$$

51. The Board of Trade and Canadian rules provide that for chain-riveted joints having each alternate rivet omitted in the outer row, or in the inner and outer rows, the distance between those rows of rivets having the larger and smaller number of rivets should be calculated by the rule given in the previous article. If the calculated value is less than twice the rivet diameter, the distance between the rows of rivets must be made at least twice the rivet diameter, and preferably $\frac{1}{2}$ inch more. It is to be observed that the greatest pitch is to be used in applying the rule referred to.

52. For a triple-chain-riveted butt joint in which every alternate rivet is omitted in the outer row, the distance between the inner and middle row of rivets, according to the Board of Trade and Canadian rules, must not be less than twice the rivet diameter, and preferably should be $\frac{1}{2}$ inch more.

53. For a triple-zigzag-riveted butt joint in which every alternate rivet is omitted in the outer row, the distance between the inner and middle row of rivets, under Board of Trade and Canadian rules, is found as follows:

Rule.—*Multiply the sum of 11 times the pitch and 8 times the rivet diameter by the sum of the pitch and 8 times the rivet diameter. Extract the square root of the product and divide the root by 20.*

Or,

$$V_1 = \frac{\sqrt{(11p + 8d)(p + 8d)}}{20}$$

EXAMPLE.—What should be the distance between the inner and middle row of rivets in a triple-zigzag-riveted butt joint in which every alternate rivet is omitted in the outer row, if the rivets are $1\frac{1}{2}$ inches in diameter and have a pitch of 9 inches in the outer row?

SOLUTION.—Applying the rule,

$$V_1 = \frac{\sqrt{(11 \times 9 + 8 \times 1\frac{1}{2}) \times (9 + 8 \times 1\frac{1}{2})}}{20} = 2.414 \text{ in. Ans.}$$

54. In a double-zigzag-riveted butt joint with every alternate rivet omitted in the outer row, and in a triple-zigzag-riveted lap joint and butt joint with every alternate rivet omitted in the inner and outer rows, the distance between the rows of rivets, according to the Board of Trade and Canadian rules, is to be found by the rule given in this article. The same rule is also used for finding the distance between the middle and outer rows of rivets in a triple-zigzag-riveted butt joint where every alternate rivet in the outer row has been omitted.

Rule.—*Multiply the sum of $\frac{11}{20}$ times the greatest pitch and the rivet diameter by the sum of $\frac{1}{20}$ times the greatest pitch and the rivet diameter. Extract the square root of the product.*

Or, $$V = \sqrt{(\tfrac{11}{20}p + d)(\tfrac{1}{20}p + d)}$$

EXAMPLE.—In a triple-riveted lap joint having every alternate rivet omitted in the inner and outer rows, the greatest pitch of the rivets is 8 inches and their diameter $1\frac{1}{4}$ inches; what should be the distance between the rows of rivets?

SOLUTION.—Applying the rule,

$$V = \sqrt{(\tfrac{11}{20} \times 8 + 1\tfrac{1}{4}) \times (\tfrac{1}{20} \times 8 + 1\tfrac{1}{4})} = 3.05 \text{ in. Ans.}$$

55. Rules for Diagonal Pitch.—The American rules specify for double-zigzag-riveted lap joints, and the Board of Trade and Canadian rules, in addition to this kind of joint, specify for triple-zigzag-riveted and quadruple-zigzag-riveted lap joints, and for double-zigzag-riveted and triple-zigzag-riveted butt joints, it being understood that joints in which alternate rivets are omitted in any row are not referred to, that the diagonal pitch of the rivets is to be found as follows:

Rule.—*To 6 times the pitch add 4 times the rivet diameter. Divide the sum by 10.*

Or, $$p_d = \frac{6p + 4d}{10}$$

EXAMPLE.—In a double-zigzag-riveted lap joint the rivets are $\frac{3}{4}$ inch in diameter and have a pitch of $2\frac{1}{2}$ inches; what should the diagonal pitch be?

SOLUTION.—Applying the rule,

$$p_d = \frac{6 \times 2\frac{1}{2} + 4 \times \frac{3}{4}}{10} = 1.8 \text{ in. Ans.}$$

56. For triple-zigzag-riveted lap joints having every alternate rivet omitted in the outer rows, and butt joints having alternate rivets omitted in the inner and outer rows, and for a double-zigzag-riveted butt joint having every alternate rivet omitted in the outer row, the Board of Trade and Canadian rules give the rule presented in this article for finding the diagonal pitch. The same rule is also applied to finding the diagonal pitch between the middle and outer rows of rivets in a triple-zigzag-riveted butt joint having every alternate rivet omitted in the outer row.

Rule.—*To $\frac{3}{10}$ times the greatest pitch add the diameter of the rivet.*

Or,
$$p_d = \tfrac{3}{10}p + d$$

EXAMPLE.—In a triple-zigzag-riveted butt joint in which every alternate rivet has been omitted in the outer row, the rivets are $1\frac{1}{4}$ inches in diameter and the pitch in the outer row is $7\frac{1}{2}$ inches. Find the diagonal pitch between the middle and outer rows of rivets.

SOLUTION.—Applying the rule,
$$p_d = \tfrac{3}{10} \times 7\tfrac{1}{2} + 1\tfrac{1}{4} = 3.5 \text{ in. Ans.}$$

57. The Board of Trade and Canadian rules specify that the diagonal pitch between the inner and middle row of rivets in a triple-zigzag-riveted butt joint having every alternate rivet omitted in the outer row, shall be found as follows:

Rule.—*To 3 times the greatest pitch add 4 times the rivet diameter and divide the sum by 10.*

Or,
$$P_d = \frac{3p + 4d}{10}$$

EXAMPLE.—Using the values given in the example in Art. **56**, what should be the diagonal pitch between the inner and middle row of rivets?

SOLUTION.— $p = 7\frac{1}{2}$ in.; $d = 1\frac{1}{4}$ in. Applying the rule,
$$P_d = \frac{3 \times 7\frac{1}{2} + 4 \times 1\frac{1}{4}}{10} = 2.75 \text{ in. Ans.}$$

EXAMPLES FOR PRACTICE

1. Under American rules, what should be the pitch of the rivets in a double-zigzag-riveted lap joint, iron plate and iron rivets, if the rivets are 1 inch in diameter and the plate is $\frac{11}{16}$ inch thick?
Ans. 3.285 in.

2. Under American rules, what should be the pitch of the rivets in a double-chain-riveted lap joint, steel plate and steel rivets, if the rivets are 1 inch in diameter and the plate is $\frac{5}{8}$ inch thick?
Ans. 3.064 in.

3. Under American rules, what is the maximum pitch permissible in the joint in example 2? Ans. 3.26 in.

4. Under Board of Trade and Canadian rules, what is the pitch of the rivets in a triple-riveted lap joint, iron plate and iron rivets, the plate being $\frac{3}{4}$ inch thick and the rivets 1 inch in diameter?
Ans. 4.14 in.

5. Under Board of Trade and Canadian rules, what is the maximum pitch permissible in the joint in example 4? Ans. 4.227 in.

6. Under American rules, what will be the rivet diameter for a steel plate $\frac{13}{16}$ inch thick, double-riveted lap joint, the rivets being steel? Ans. $1\frac{3}{16}$ in.

7. If the rivets are $1\frac{3}{8}$ inches in diameter, how far must their center be from the edge of the plate? Ans. $2\frac{1}{16}$ in.

8. What should be the distance between the rows of rivets in a double-zigzag-riveted lap joint if the rivets are 1 inch in diameter and the pitch is 3 inches? Ans. 1.609 in.

9. In a triple-zigzag-riveted butt joint having every alternate rivet omitted in the outer row, what should be the distance between the middle and inner row of rivets if the rivets are 1 inch in diameter and have a pitch of 6 inches in the outer row? Ans. 1.609 in.

10. What should be the distance between the middle and outer rows of rivets in the joint in example 9? Ans. 2.364 in.

11. What should be the diagonal pitch in an ordinary double-zigzag-riveted lap joint if the rivets are 1 inch in diameter and the pitch is 3 inches? Ans. 2.2 in.

12. In a double-zigzag-riveted butt joint every alternate rivet is omitted in the outer row; the pitch in the outer row being 5 inches and the rivets $\frac{15}{16}$ inch in diameter, what should the diagonal pitch be?
Ans. $2\frac{7}{16}$ in.

13. Find the diagonal pitch between the inner and middle row of rivets in a triple-zigzag-riveted butt joint having every alternate rivet omitted in the outer row, the rivets being $1\frac{3}{8}$ inches in diameter and 8 inches pitch in the outer row. Ans. 2.95 in.

EFFICIENCY OF RIVETED JOINTS

58. The ratio between the strength of the plate and the strength of the joint is called the **efficiency of the joint,** and is expressed as a percentage of the strength of the solid plate.

In order to determine the efficiency of the joint, its resistance must be computed for each of the different ways in which it may fail; the lowest efficiency found will be the efficiency of the joint.

A riveted joint may fail in several ways: (1) The plate may break along the rivet holes. (2) The rivets may shear off. (3) The plate may shear out in front of the rivet. (4) The plate may crush in front of the rivet. (5) In zigzag-riveted joints, the plate may break diagonally between the rivet holes. With joints having the proportions given by the Board of Trade, Canadian, and American rules, the liability of a joint failing in the way described under (3), (4), and (5) is extremely remote.

59. The American rules do not contain any formula for finding the efficiency of a joint; the Board of Trade and Canadian rules present the formulas here given. In these formulas,

p = greatest pitch of rivets, in inches;
d = diameter of rivet, in inches;
T = thickness of plate, in inches;
S_1 = tensile strength of steel plate, in gross tons (2,240 pounds);
n = number of rivets in one pitch;
r = percentage of plate left between holes in greatest pitch;
R = percentage of value of rivet section;
R_1 = percentage of combined plate and rivet section.

60. For failure of the joint by breaking of the plate, for iron plate and iron rivets or steel plate and steel or iron rivets, the Board of Trade and Canadian rules state that the efficiency is to be calculated as follows, calling it the percentage of plate left between holes in greatest pitch:

Rule.—*Multiply 100 by the difference between the greatest pitch and the rivet diameter, in inches. Divide the product by the greatest pitch, in inches.*

Or,
$$r = \frac{100\,(p - d)}{p}$$

EXAMPLE.—What is the efficiency of a double-riveted lap joint, plate and rivets being steel, calculated for failure by breaking of the plate, when the pitch of the rivets is 3 inches and their diameter is 1 inch?

SOLUTION.—Applying the rule,
$$r = \frac{100 \times (3 - 1)}{3} = 66.67 \text{ per cent. Ans.}$$

61. For failure of the joint by shearing of the rivets, for iron plates and iron rivets, the Board of Trade and Canadian rules state that the efficiency is to be calculated as follows, calling the efficiency the **percentage of value of rivet section:**

Rule.—*Multiply 100 by the square of the rivet diameter, by .7854, by the number of rivets in one pitch, and by 1.75 if the rivets are in double shear. Divide the product by the product of the greatest pitch and plate thickness, in inches.*

Or, for single shear,
$$R = \frac{100\,d^2\,.7854\,n}{p\,T}$$
and for double shear,
$$R = \frac{100\,d^2\,.7854\,n\,1.75}{p\,T}$$

The rule just given assumes that the shearing strength and tensile strength of wrought iron are equal. This assumption is incorrect, as the shearing strength of wrought iron is in reality somewhat less than its tensile strength. The rule must be used, however, by candidates for marine engineer's license.

EXAMPLE.—Given a double-riveted lap joint, iron plate and iron rivets, the plate being $\frac{3}{4}$ inch thick, the rivets $1\frac{1}{16}$ inches in diameter, and the pitch 3.426 inches; what is the efficiency calculated for failure by shearing of the rivets?

SOLUTION.— $n = 2$. Applying the rule, remembering that the rivets are in single shear,

$$R = \frac{100 \times (1\frac{1}{16})^2 \times .7854 \times 2}{3.426 \times \frac{3}{4}} = 69 \text{ per cent., nearly. Ans.}$$

62. For failure by shearing of the rivets, with steel plates and steel rivets, the Board of Trade prescribes the rule given in this article. The Canadian rule differs slightly in the value of one of the factors. The result is called the percentage of value of rivet section.

Rule.—*Multiply 100 by 23, by the square of the rivet diameter, by .7854, by the number of rivets in one pitch, by 1.75 if the rivets are in double shear, and by the factor of safety demanded by the construction of the joint. Divide the product by the product of 4.5 (4.25 under Canadian Rules) and the tensile strength of the plate, in gross tons, and the greatest pitch, in inches, and the plate thickness, in inches.*

Or, for single shear (Board of Trade),

$$R = \frac{100 \times 23\, d^2\, .7854\, n\, F}{4.5\, S_1\, p\, T}$$

and for double shear (Board of Trade),

$$R = \frac{100 \times 23\, d^2\, .7854\, n\, 1.75\, F}{4.5\, S_1\, p\, T}$$

and for single shear (Canada),

$$R = \frac{100 \times 23\, d^2\, .7854\, n\, F}{4.25\, S_1\, p\, T}$$

and for double shear (Canada),

$$R = \frac{100 \times 23\, d^2\, .7854\, n\, 1.75\, F}{4.25\, S_1\, p\, T}$$

EXAMPLE.—In a triple-riveted, double-butt-strap joint having alternate rivets omitted in the outer row, the rivets are $1\frac{1}{8}$ inches in diameter and pitched $7\frac{1}{2}$ inches in the outer row and $3\frac{3}{4}$ inches in the middle and inner rows. The plate is 1 inch thick and has a tensile strength of 30 gross tons per square inch of section. The factor of safety being 5, under Board of Trade rules, what is the efficiency of

the joint calculated for failure by shearing of the rivets? Plate and rivets are steel.

SOLUTION.—In this case, $n = 5$. Applying the rule, remembering that the rivets are in double shear,

$$R = \frac{100 \times 23 \times (1\frac{1}{8})^2 \times .7854 \times 5 \times 1.75 \times 5}{4.5 \times 30 \times 7\frac{1}{2} \times 1} = 98.79 \text{ per cent. Ans.}$$

63. The Board of Trade rules, for failure by shearing of the rivets, with iron rivets and steel plate, specify that the calculation for efficiency is to be made as follows, calling the efficiency the percentage of value of rivet section:

Rule.—*Multiply 100 by 17.5, by the square of the rivet diameter, in inches, by .7854, by the number of rivets in one pitch, by 1.75 if the rivets are in double shear, and by the factor of safety demanded by the construction. Divide the product by the product of 4.5, the tensile strength of the plate, in gross tons, the greatest pitch of the rivets, and the plate thickness, in inches.*

Or, for single shear,

$$R = \frac{100 \times 17.5\, d^2\, .7854\, n\, F}{4.5\, S_1\, p\, T}$$

and for double shear,

$$R = \frac{100 \times 17.5\, d^2\, .7854\, n\, 1.75\, F}{4.5\, S_1\, p\, T}$$

EXAMPLE.—In a double-riveted lap joint, the steel plate is $\frac{1}{2}$ inch thick and has a tensile strength of 28 gross tons. The iron rivets are $\frac{15}{16}$ inch diameter and have a pitch of 2.9 inches. The factor of safety being 5.2, what is the efficiency of the joint calculated for failure by shearing of the rivets?

SOLUTION.—In this case, $n = 2$. Remembering that the rivets are in single shear, and applying the rule,

$$R = \frac{100 \times 17.5 \times (\frac{15}{16})^2 \times .7854 \times 2 \times 5.2}{4.5 \times 28 \times 2.9 \times \frac{1}{2}} = 68.76 \text{ per cent. Ans.}$$

64. The Canadian rules, for failure by shearing of the rivets, with iron rivets and steel plates, specify that the calculation for efficiency is to be made as follows, calling the efficiency the percentage of value of rivet section:

Rule.—*Multiply 100 by 8, by the square of the rivet diameter, by .7854, by the number of rivets in one pitch, by 1.75 if*

the rivets are in double shear, and by the factor of safety demanded by the construction. Divide the product by the product of 4.25, and 13, and the greatest pitch, in inches, and the plate thickness, in inches.

Or, for single shear,

$$R = \frac{100 \times 8\, d^2\, .7854\, n\, F}{4.25 \times 13\, p\, T}$$

and for double shear,

$$R = \frac{100 \times 8\, d^2\, .7854\, n\, 1.75\, F}{4.25 \times 13\, p\, T}$$

EXAMPLE.—Taking the example given in Art. **63**, calculate the efficiency of the joint by the Canadian rule.

SOLUTION.—Applying the rule,

$$R = \frac{100 \times 8 \times (\frac{15}{16})^2 \times .7854 \times 2 \times 5.2}{4.25 \times 13 \times 2.9 \times \frac{1}{2}} = 71.69 \text{ per cent. Ans.}$$

65. Riveted joints in which every alternate rivet has been omitted in the outer row, or in the inner and outer rows, may fail by breaking of the plate in the row or rows having the greatest number of rivets and the shearing of rivets. For this manner of failure, the Board of Trade and Canadian rules state that the efficiency is to be calculated as follows, calling it the **percentage of combined plate and rivet section:**

Rule.—*Multiply 100 by the difference between the greatest pitch and twice the rivet diameter. Divide the product by the greatest pitch. To the quotient add the quotient obtained by dividing the efficiency of the joint calculated for failure by shearing of the rivets, by the number of rivets in one pitch.*

Or,
$$R_1 = \frac{100\,(p - 2d)}{p} + \frac{R}{n}$$

EXAMPLE.—In a triple-zigzag-riveted joint having double butt straps, every alternate rivet is omitted in the outer row. The rivets are $1\frac{1}{8}$ inches in diameter and pitched $7\frac{1}{2}$ inches in the outer row. Plate and rivets are steel, the plate being 1 inch thick and having a tensile strength of 28 gross tons. The factor of safety, under Board of Trade rules, is 5. Calculate the percentage of combined plate and rivet section.

SOLUTION.—The percentage of value of the rivet section has to be calculated first. For the case given, the rule in Art. **62** applies.

Remembering that $n = 5$, and that the rivets are in double shear,

$$R = \frac{100 \times 23 \times (1\tfrac{1}{8})^2 \times .7854 \times 5 \times 1.75 \times 5}{4.5 \times 28 \times 7\tfrac{1}{2} \times 1} = 105.84 \text{ per cent.}$$

Applying the rule in this article,

$$R_1 = \frac{100 \times (7\tfrac{1}{2} - 2 \times 1\tfrac{1}{8})}{7\tfrac{1}{2}} + \frac{105.84}{5} = 91.17 \text{ per cent. Ans.}$$

66. The application of the rules will now be shown.

EXAMPLE 1.—Calculate the efficiency of a double-chain-riveted butt joint with single butt strap, plate and rivets wrought iron; the plate is $\frac{11}{16}$ inch thick, the rivets are 1 inch diameter, and the pitch is $3\frac{1}{4}$ inches.

SOLUTION.—For the joint given, $n = 2$. The rivets are in single shear. Applying the rule in Art. **60**,

$$r = \frac{100 \times (3\tfrac{1}{4} - 1)}{3\tfrac{1}{4}} = 69.23 \text{ per cent.}$$

Applying the rule in Art. **61**,

$$R = \frac{100 \times 1^2 \times .7854 \times 2}{3\tfrac{1}{4} \times \tfrac{11}{16}} = 70.3 \text{ per cent.}$$

Then, efficiency of joint is 69.23 per cent. Ans.

EXAMPLE 2.—Calculate the efficiency of a single-riveted lap joint, plate and rivets of steel; the plate is $\frac{9}{16}$ inch thick and has a tensile strength of 28 gross tons. The rivets are 1 inch in diameter and have a pitch of $2\frac{1}{8}$ inches. Use Board of Trade rules and a factor of safety of 5.5.

SOLUTION.—For this case, $n = 1$; the rivets are in single shear. Applying the rule in Art. **60**,

$$r = \frac{100 \times (2\tfrac{1}{8} - 1)}{2\tfrac{1}{8}} = 52.94 \text{ per cent.}$$

Applying the rule in Art. **62**,

$$R = \frac{100 \times 23 \times 1^2 \times .7854 \times 1 \times 5.5}{4.5 \times 28 \times 2\tfrac{1}{8} \times \tfrac{9}{16}} = 65.93 \text{ per cent.}$$

Efficiency of joint is 52.94 per cent. Ans.

EXAMPLE 3.—Calculate the efficiency of a triple-riveted, double-butt-strap joint in which every alternate rivet has been omitted in the outer row. Plate and rivets are steel. The plate is 1 inch thick and has a tensile strength of 30 gross tons. The rivets are $1\frac{1}{4}$ inches in diameter and are pitched $7\frac{3}{4}$ inches in the outer row and $3\frac{7}{8}$ inches in the inner row. The factor of safety is 5. Use Canadian rules.

SOLUTION.—For this case, $n = 5$; the rivets are in double shear. Applying the rule in Art. **60**,

$$r = \frac{100 \times (7\tfrac{3}{4} - 1\tfrac{1}{4})}{7\tfrac{3}{4}} = 83.87 \text{ per cent.}$$

Applying the rule in Art. **62**,

$$R = \frac{100 \times 23 \times (1\frac{1}{4})^2 \times .7854 \times 5 \times 1.75 \times 5}{4.25 \times 30 \times 7\frac{3}{4} \times 1} = 124.97 \text{ per cent.}$$

Applying the rule in Art. **65**,

$$R_1 = \frac{100 \times (7\frac{3}{4} - 2 \times 1\frac{1}{4})}{7\frac{3}{4}} + \frac{124.97}{5} = 92.73 \text{ per cent.}$$

Efficiency of joint is 83.87 per cent. Ans.

EXAMPLE 4.—Calculate, by Canadian rules, the efficiency of a triple-riveted, single-butt-strap joint in which the plate is of steel and $\frac{1}{2}$ inch thick. The iron rivets are $\frac{7}{8}$ inch in diameter and $3\frac{1}{4}$ inches pitch. The factor of safety is 4.9.

SOLUTION.—For this case, $n = 3$; the rivets are in single shear. Applying the rule in Art. **60**,

$$r = \frac{100 \times (3\frac{1}{4} - \frac{7}{8})}{3\frac{1}{4}} = 73.08 \text{ per cent.}$$

Applying the rule in Art. **64**,

$$R = \frac{100 \times 8 \times (\frac{7}{8})^2 \times .7854 \times 3 \times 4.9}{4.25 \times 13 \times 3\frac{1}{4} \times \frac{1}{2}} = 78.76 \text{ per cent.}$$

Efficiency of joint is 73.08 per cent. Ans.

EXAMPLES FOR PRACTICE

1. What is the percentage of plate left between the rivets in the greatest pitch in a double-riveted, double-butt-strap joint having rivets $\frac{3}{4}$ inch in diameter and 2.8 inches pitch? Ans. 73.21 per cent.

2. What is the percentage of value of the rivet section of the joint in example 1, plate and rivets being iron and the plate being $\frac{9}{16}$ inch thick? Ans. 98.17 per cent.

3. Given a triple-zigzag-riveted lap joint, steel plate and steel rivets. The plate is $\frac{1}{2}$ inch thick, and has a tensile strength of 28 gross tons. The rivets are $\frac{13}{16}$ inch in diameter and their pitch is 3.3 inches. The factor of safety being 5, find the percentage of value of the rivet section under Canadian rules. Ans. 91.1 per cent.

4. If the rivets in example 3 were wrought iron, what would the percentage of value of the rivet section be under Canadian rules, assuming them to be $\frac{7}{8}$ inch in diameter? Ans. 79.15 per cent.

5. In a triple-chain-riveted, double-butt-strap joint, alternate rivets are omitted in the inner and outer rows. The rivets are $1\frac{1}{4}$ inches in diameter and pitched $7\frac{1}{4}$ inches in the inner and outer rows. Plate and rivets are steel, the plate being $\frac{7}{8}$ inch thick and having a tensile strength of 27 gross tons. The factor of safety being 5.2, calculate the percentage of the combined plate and rivet section under Board of Trade rules. Ans. 98.84 per cent.

MARINE-BOILER INSPECTION

(PART 2)

AMERICAN, BRITISH, AND CANADIAN RULES

OPENINGS IN BOILERS

MANHOLES

1. The American rules state: "All manholes for the shell of boilers over 40 inches in diameter, when practicable for use, shall have an opening not less than 10 by 16 or 11 by 15 inches in the clear, except that boilers 40 inches diameter of shell and under shall have an opening of not less than 9 by 15 inches in the clear in manholes: Provided, That manhole opening in front head of externally fired boilers, under the flues, so required by section 4434, Revised Statutes of the United States, shall be of dimensions not less than 8 by 12 inches in the clear."

2. Neither the Board of Trade nor the Canadian rules specify a minimum size of manholes; both specify that manhole and handhole openings in the shell of cylindrical boilers should have their shorter axes placed longitudinally. The Board of Trade rules prohibit the use of cast-iron manhole plates; the American rules permit cast iron to be used for this purpose, stating: "Provided, however, that the material shall be of the best grade and of suitable thickness and uniform section for the pressure allowed on boilers."

REINFORCEMENT OF OPENINGS

3. The American rules state: "When holes exceeding 6 inches in diameter are cut in boilers for pipe connections, man- and handhole plates, such holes shall be reinforced, either on the inside or outside of boiler, with reinforcing plates, which shall be securely riveted or properly fastened to the boiler, such reinforcing material to be rings of the same kind and quality as the material reinforced, and of sufficient width and thickness of material to equal the amount of material cut from such boilers, in flat surfaces; and where such opening is made in the circumferential plates of such boilers, the reinforcing ring shall have an area of at least one-half the area of material there would be in a line drawn across such opening parallel with the longitudinal seams of such portion of the boiler. On boilers carrying 75 pounds or less steam pressure, a cast-iron stop-valve, properly flanged, may be used as a reinforcement to such opening. When holes are cut in any flat surface of such boilers and such holes are flanged inwardly to a depth of not less than $1\frac{1}{2}$ inches, measuring from the outer surface, the reinforcement rings may be dispensed with.

4. The Board of Trade rules state: "Compensating rings of at least the same sectional area as the plate cut out, should be fitted around all manholes and openings, and in no case should the rings be less in thickness than the plates to which they are attached.

"It is very desirable that the compensating rings around openings in flat surfaces should be made of **L** or **T** iron."

5. The Canadian rules state: "Manhole openings must be stiffened with compensating plates or rings of at least the same effective sectional area as the plate cut out, and in no case shall such plates or rings be of less thickness than the plate to which they are attached, nor the attachment of less strength than the plate or ring. All openings in the shells of boilers should have their short axis placed longitudinally, and if not so placed must have compensating

plates or rings, and attachments, equal to twice the effective sectional area of the plate cut out."

When cast-iron frames are fastened around manholes in steamboat boilers, a compensating ring in addition must be provided.

Mud-holes or handholes should not be placed in boiler shells of a greater short diameter than 5 inches, unless provided with reinforcing plates, and if on the cylindrical shell of a boiler, their short diameter should be in a line with the axis of the shell.

FLAT SURFACES AND STAYING

STRENGTH OF FLAT SURFACES

6. The American rules provide that the working pressure on flat surfaces supported by stays shall be determined by the rule given in this article, defining as a flat surface any stayed surface formed to a curve having a radius over 21 inches.

Rule.—*Multiply the constant corresponding to the conditions by the square of the plate thickness, in sixteenths of an inch, and divide the product by the square of the greatest pitch of the stays, in inches.*

Or,

$$P_w = \frac{C\,T^2}{P^2}$$

in which P_w = working pressure, in pounds per square inch;
T = thickness of plate, in sixteenths of an inch;
P = greatest pitch of stays;
C = 112 for ordinary riveted screw stays and plates $\frac{7}{16}$ inch thick or under;
C = 120 for ordinary riveted screw stays and plates over $\frac{7}{16}$ inch thick;
C = 140 for plates fitted with stays having one nut on the inside and one nut on the outside of the plate;

C = 160 for plates fitted with washers having at least half of the thickness of the plate and a diameter of at least half of the greatest pitch of the stays, riveted to the outside of the plates, and having one nut inside of the plate and one nut outside of the washer: T will then equal 80 per cent. of the combined thickness of the plate and washer;

C = 200 for plates fitted with doubling plates which have a thickness equal to at least half of the thickness of the plate reinforced and covering the full area braced (up to the curvature of the flange, if any), riveted to either the inside or outside of the plate, and stays having one nut outside and one inside of the plates: provided, that the washers or doubling plates are riveted to the plates with rivets spaced and of sufficient sectional area as provided for stays and flat surfaces, the pitch to be determined by the thickness of the washer or doubling plate: T will then equal 80 per cent. of the combined thickness of the two plates;

C = 200 for plates fitted with tees or angle bars having a thickness of at least two-thirds the thickness of plate and depth of webs at least one-quarter the greatest pitch of the stays, and riveted on the inside of the plates, and stays having one nut inside bearing on washers fitted to the edges of the webs that are at right angles to the plate: T will then equal 80 per cent. of the combined thickness of web and plate.

For the cases where C = 160 and 200, T may be found as follows: Assume that the combined thickness is $1\frac{1}{8}$ inches. Then, 80 per cent. of this is $1\frac{1}{8} \times .8 = .9$ inch. This value must be expressed in sixteenths, which may be found by

multiplying the value (expressed decimally) by 16. Thus, .9 inch = .9 × 16 = 14.4 sixteenths of an inch.

EXAMPLE 1.—What working pressure will be allowed on a flat plate $\frac{3}{8}$ inch thick, fitted with screw stays having a pitch of 7 inches one way and $6\frac{1}{2}$ inches the other way?

SOLUTION.—Since $\frac{3}{8} = \frac{6}{16}$, $T = 6$. For this case, $C = 112$. Applying the rule,

$$P_w = \frac{112 \times 6^2}{7^2} = 82.29 \text{ lb. per sq. in. Ans.}$$

EXAMPLE 2.—A plate $\frac{3}{4}$ inch thick is supported by screw stays having a pitch of 10 inches; what working pressure will be allowed on this plate?

SOLUTION.—Since $\frac{3}{4} = \frac{12}{16}$, $T = 12$. For this case, $C = 120$. Applying the rule,

$$P_w = \frac{120 \times 12^2}{10^2} = 172.8 \text{ lb. per sq. in. Ans.}$$

EXAMPLE 3.—What working pressure will be allowed on a plate $\frac{13}{16}$ inch thick that is supported by stayrods 14 inches from center to center, the stayrods being fitted with one nut on the inside and one nut on the outside of the plate?

SOLUTION.— $T = 13$. For this case, $C = 140$. Applying the rule,

$$P_w = \frac{140 \times 13^2}{14^2} = 120.71 \text{ lb. per sq. in. Ans.}$$

EXAMPLE 4.—A plate $\frac{5}{8}$ inch thick is braced by stayrods spaced 14 inches center to center and is reinforced by washers 8 inches in diameter and $\frac{3}{8}$ inch thick, properly riveted; the stayrods have one nut inside the plate and one nut outside the washer. What working pressure will be allowed on the plate?

SOLUTION.— $\frac{5}{8} + \frac{3}{8} = 1$ in. 80 per cent. of this is $1 \times .8 = .8$ in. Reduced to sixteenths, this is $.8 \times 16 = 12.8$. For this case, $C = 160$. Applying the rule,

$$P_w = \frac{160 \times 12.8^2}{14^2} = 133.75 \text{ lb. per sq. in. Ans.}$$

EXAMPLE 5.—A plate $\frac{3}{4}$ inch thick is reinforced with a doubling plate $\frac{1}{2}$ inch thick and properly riveted; the plate is braced by stayrods spaced 15 inches center to center and each is supplied with a nut on the inside and outside of the plate. Find the allowable working pressure.

SOLUTION.— $\frac{3}{4} + \frac{1}{2} = 1\frac{1}{4}$. 80 per cent. of this is $1\frac{1}{4} \times .8 = 1$ in., whence $T = 16$. For this case, $C = 200$. Applying the rule,

$$P_w = \frac{200 \times 16^2}{15^2} = 227.56 \text{ lb. per sq. in. Ans.}$$

7. The American rules provide that the maximum pitch of stays, measured from center to center, must not exceed $10\frac{1}{2}$ inches when the plates are exposed to the impact of the heat or flame, and 18 inches in all other cases.

8. In applying any one of the rules relating to the strength of flat surfaces it must be remembered that it presupposes that the stress per square inch of section does not exceed the lawful limit. Should the stress be more, the pressure must be reduced to suit the size of the stays.

9. The Board of Trade, and also the Canadian, rule for the working pressure allowable on flat surfaces is as follows:

Rule.—*Multiply the constant corresponding to the circumstances by the square of the sum of the plate thickness, in sixteenths of an inch, and 1. Divide the product by the difference between the number of square inches of surface supported and 6.*

Or,
$$B = \frac{C(T+1)^2}{S-6}$$

in which B = working pressure, in pounds per square inch;

T = numerator of fraction expressing plate thickness, in sixteenths of an inch;

S = surface supported, in square inches;

C = 192 under Board of Trade and 160 under Canadian rules, when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts on both sides of the plates and doubling strips not less in width than two-thirds the pitch of the stays and of the thickness of the plates, are securely riveted to the outside of the plates they cover;

C = 168 under Board of Trade and 150 under Canadian rules, when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts on both sides of the plates, and with washers not less in diameter than two-thirds the pitch of the stays and of the same thickness as the plates, securely riveted to the outside of the plates they cover;

C = 132 under Board of Trade and 100 under Canadian rules, when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts on both sides of the plates, and with washers outside the plates at least three times the diameter of the stay, and two-thirds the thickness of the plates they cover;

C = 120 under Board of Trade and 90 under Canadian rules, when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts on both sides of the plates;

C = 90 under Board of Trade rules, when tube plates are not exposed to the direct impact of heat or flame, and the stays are fitted with nuts;

C = 70 under Board of Trade rules, when tube plates are not exposed to the direct impact of heat or flame, and the stay-tubes are screwed and expanded;

C = 70 under both Board of Trade and Canadian rules, when the plates are not exposed to the impact of heat or flame, and the stays are screwed into the plates and riveted over;

C = 60 under both Board of Trade and Canadian rules, when the plates are exposed to the impact of heat or flame, with steam in contact with the plates, and the stays are fitted with nuts and washers, the latter being at least three times the diameter of the stay, and two-thirds the thickness of the plates they cover;

C = 54 under both Board of Trade and Canadian rules, when the plates are exposed to the impact of heat or flame, with steam in contact with the plates, and the stays are fitted with nuts only;

$C = 80$ under both Board of Trade and Canadian rules, when the plates are exposed to the impact of heat or flame, with water in contact with the plates, and the stays are screwed into the plates and fitted with nuts;

$C = 60$ under both Board of Trade and Canadian rules, when the plates are exposed to the impact of heat or flame, with water in contact with the plates, and the stays are screwed into the plates, and have the ends riveted over to form substantial heads;

$C = 36$ under both Board of Trade and Canadian rules, when the plates are exposed to the impact of heat or flame, with steam in contact with the plates, with the stays screwed into the plates, and having the ends riveted over to form substantial heads.

Both the Board of Trade and Canadian rules state: "When the riveted ends of screwed stays are much worn, or when the nuts are burned, the constants should be reduced, but the surveyor (inspector, in Canada) must act according to the circumstances that present themselves at the time of the survey (inspection, in Canada), and it is expected that in cases where the riveted ends of screwed stays in the combustion boxes and furnaces are found in this state it will be often necessary to reduce the constant 60 to about 36."

All the values of C given in this article are for iron boilers. For steel boilers in which the material complies with the specifications, both the Board of Trade and Canadian rules state: "If flanged plates and plates exposed to flame comply with the foregoing conditions, the constants in the Rules for iron boilers may be increased as follows:

"The constants for flat surfaces when they are supported by stays screwed into the plate and riveted, 10 per cent.

"The constants for flat surfaces when they are supported by stays screwed into the plate and nutted, or when the stays are nutted in the steam space, 25 per cent. This is also applicable to the constants for flat surfaces stiffened by

riveted washers or doubling strips, and supported by nutted stays."

EXAMPLE 1.—What working pressure, under Canadian rules, will be allowed on a combustion-chamber back plate $\frac{5}{8}$ inch thick, made of wrought iron, and stayed by screwed staybolts having a pitch of $6\frac{1}{2}$ inches and fitted with nuts?

SOLUTION.— $\frac{5}{8} = \frac{10}{16}$, or $T = 10$.

$$S = 6\tfrac{1}{2} \times 6\tfrac{1}{2} = 42.25 \text{ sq. in.}$$

The plate being in contact with water and exposed to the direct impact of the flame, and the screwed staybolts having nuts, $C = 80$. Applying the rule,

$$B = \frac{80 \times (10 + 1)^2}{42.25 - 6} = 267 \text{ lb. per sq. in. Ans.}$$

EXAMPLE 2.—What working pressure, under Board of Trade rules, will be allowed on the back head of a single-ended Scotch boiler built of wrought iron, the head being 1 inch thick, and the stayrods having a pitch of 15 inches and being fitted with nuts on each side of the plate, but no washers?

SOLUTION.— $1 = \frac{16}{16}$, or $T = 16$.

$$S = 15 \times 15 = 225 \text{ sq. in.}$$

The back head of a single-ended Scotch boiler not being in contact with flame, $C = 120$ for this case. Applying the rule,

$$B = \frac{120 \times (16 + 1)^2}{225 - 6} = 158.36 \text{ lb. per sq. in. Ans.}$$

EXAMPLE 3.—If the boiler in example 2 were built of steel having passed inspection, what pressure would be allowed on the back head?

SOLUTION.—In this case, the constant 120 may be increased 25 per cent. Then,

$$C = 120 \times 1.25 = 150$$

Applying the rule,

$$B = \frac{150 \times (16 + 1)^2}{225 - 6} = 197.95 \text{ lb. per sq. in. Ans.}$$

10. When plates not exposed to the impact of the heat or flame are reinforced by doubling plates riveted to them, said doubling plates covering the whole of the flat surface, the working pressure, according to the Board of Trade rules, is to be found as follows:

Rule.—*Multiply the constant applicable to the case* (*as given in Art.* **9**) *by the square of the sum of the plate thickness, in sixteenths of an inch, and 1. Add to the product, the product of the constant and the square of the sum of the doubling-plate*

thickness, in sixteenths of an inch, and 1. Divide the sum by the number of square inches of surface supported diminished by 6.

Or, $$B = \frac{C(T+1)^2 + C(T_1+1)^2}{S-6}$$

in which T_1 is the numerator of the fraction expressing the thickness of the doubling plate, in sixteenths of an inch, and the other letters have the same meaning as in Art. **9.**

EXAMPLE 1.—The front head of a firebox boiler is of wrought iron $\frac{1}{2}$ inch thick and reinforced with a doubling plate $\frac{3}{8}$ inch thick; it is stayed with stayrods having nuts inside and outside, and having a pitch of 14 inches. What working pressure will be allowed?

SOLUTION.— $\frac{1}{2} = \frac{8}{16}$, or $T = 8$. $\frac{3}{8} = \frac{6}{16}$, or $T_1 = 6$.

$$S = 14 \times 14 = 196$$

By Art. **9,** $C = 120$. Applying the rule,

$$B = \frac{120 \times (8+1)^2 + 120 \times (6+1)^2}{196-6} = 82.1 \text{ lb. per sq. in. Ans.}$$

EXAMPLE 2.—If the boiler in example 1 were built of steel, what working pressure would be allowed on the head?

SOLUTION.—The constant 120, by Art. **9,** can be increased 25 per cent. Then,

$$C = 120 \times 1.25 = 150$$

Applying the rule in Art. **10,**

$$B = \frac{150 \times (8+1)^2 + 150 \times (6+1)^2}{196-6} = 102.63 \text{ lb. per sq. in. Ans.}$$

11. The Canadian rules state that when doubling plates cover the whole of the flat surfaces, and have a thickness of not less than two-thirds or more than the thickness of the plate covered, and are substantially riveted, the working pressure on such reinforced surfaces is to be found as follows:

Rule.—*Multiply the constant 140 for iron, or 175 for steel by the square of the sum of the plate thickness and one-half the doubling-plate thickness, both expressed in sixteenths of an inch. Divide the product by the number of square inches supported.*

Or, for iron, $$B = \frac{140\left(T + \frac{t}{2}\right)^2}{S}$$

and for steel, $$B = \frac{175\left(T + \frac{t}{2}\right)^2}{S}$$

in which t is the numerator of the fraction expressing the thickness of the doubling plate, in sixteenths of an inch, and the other letters have the same meaning as in Art. **9.**

EXAMPLE.—The front head of a firebox boiler made of steel is $\frac{5}{8}$ inch thick and reinforced by a doubling plate $\frac{1}{2}$ inch thick riveted to it; the stayrods being placed 15 inches center to center, what working pressure would be allowed?

SOLUTION.— $\frac{5}{8} = \frac{10}{16}$, or $T = 10$. $\frac{1}{2} = \frac{8}{16}$, or $t = 8$.

$$S = 15 \times 15 = 225$$

Applying the rule,

$$B = \frac{175 \times (10 + \frac{8}{2})^2}{225} = 152.44 \text{ lb. per sq. in.} \quad \text{Ans.}$$

12. The Board of Trade rules state that when doubling plates do not cover the whole of the flat surfaces but are fitted between the rows of supporting stays, the working pressure allowed on such flat surfaces should be only two-thirds of that which would be allowed for similar doubling plates extending beyond and embracing the supporting stays, as found by the rule in Art. **10.**

13. The Canadian rules give the following rule for finding the working pressure on flat steel plates that are in no way exposed to the action of the fire or hot gases and are supported by stays having nuts, and washers equal in diameter to at least one-third of the pitch of the stays and a thickness of one-half the thickness of the plate:

Rule.—*Multiply 48,000 by the square of the plate thickness, in inches, and divide the product by the square of the greatest pitch of the stays, in inches.*

Or, $$P_w = \frac{48{,}000\, T^2}{P^2}$$

in which T = thickness of plate, in inches;
P = greatest pitch of stays, in inches;
P_w = working pressure, in pounds per square inch.

EXAMPLE.—A steel plate not exposed to fire or hot gases is 1 inch thick and is supported by stayrods having a pitch of 14 inches, the stayrods being fitted with washers $\frac{5}{8}$ inch thick and 5 inches in diameter; what working pressure will be allowed?

SOLUTION.—Applying the rule,

$$P_w = \frac{48,000 \times 1^2}{14^2} = 244.9 \text{ lb. per sq. in.} \quad \text{Ans.}$$

14. When tube plates are not reinforced by doubling plates between the nests of tubes, the working pressure allowable on the part of the tube plates between the nests of tubes is, under Board of Trade rules, to be calculated by the rule in Art. **9**, and when doubling plates are fitted, by the rule in Art. **10**. The value of S in those rules is to be taken as one-half the sum of the square of the horizontal pitch of the stay-tubes and the square of the vertical pitch of the stay-tubes, both pitches being in inches. The pitches are to be measured from center to center of the stay-tubes, and no deduction is permitted for any tubes in the surface included in the rectangle formed by lines joining the centers of the stay-tubes nearest the space between the nests.

EXAMPLE.—The back tube sheet of a three-furnace Scotch boiler is $\frac{7}{8}$ inch thick, and made of steel; the stay-tubes, which are fitted with nuts in the bounding rows of the nests of tubes, are placed 15 inches center to center horizontally and $9\frac{3}{4}$ inches vertically. What working pressure will be allowed on the part of the tube-sheet between the nests of tubes? The boiler has one large combustion chamber.

SOLUTION.— $S = \dfrac{15^2 + (9\frac{3}{4})^2}{2} = 160$ sq. in. nearly

By Art. **9**, $C = 80$, which may be increased 25 per cent. for steel; that is, for this case

$$C = 80 \times 1.25 = 100$$

As no doubling plate is fitted, the rule in Art. **9** applies. $\frac{7}{8} = \frac{14}{16}$, or $T = 14$. Applying the rule,

$$B = \frac{100 \times (14 + 1)^2}{160 - 6} = 146.1 \text{ lb. per sq. in.} \quad \text{Ans.}$$

15. The working pressure allowable on the part of the tube-sheet that receives the tubes is found, under Board of Trade rules, by the rule given in Art. **9**. The value of S in that rule is found for this case by multiplying together the horizontal and the vertical pitch of the stay-tubes, in inches, and subtracting therefrom the aggregate area, in square inches, of the tubes in the space bounded by four stay-tubes, plus the area of one stay-tube.

EXAMPLE.—What working pressure, under Board of Trade rules, is allowable on that part of a steel tube-sheet that contains the nest tubes if the sheet is $\frac{7}{8}$ inch thick; the stay-tubes have a horizontal pitch of 7 inches and a vertical pitch of $9\frac{3}{4}$ inches; there are five ordinary tubes $2\frac{1}{2}$ inches in outside diameter in the space bounded by four stay-tubes, and the stay-tubes are $2\frac{3}{4}$ inches in outside diameter?

SOLUTION.— $\frac{7}{8} = \frac{14}{16}$, or $T = 14$.

$$C = 80 \times 1.25 = 100$$

S is found as follows: Area of rectangle formed by lines drawn from center to center of four stay-tubes is $7 \times 9\frac{3}{4} = 68.25$ sq. in. Area of stay-tube is $(2\frac{3}{4})^2 \times .7854 = 5.94$ sq. in. Area of ordinary tubes is $5 \times (2\frac{1}{2})^2 \times .7854 = 24.54$ sq. in. Then,

$$S = 68.25 - (24.54 + 5.94) = 37.77$$

Applying the rule in Art. **9**,

$$B = \frac{100 \times (14 + 1)^2}{37.77 - 6} = 708 \text{ lb. per sq. in., nearly.} \quad \text{Ans.}$$

16. The Canadian rules do not contain any rules for finding the working pressure on the flat surface of tube plates of iron boilers, prescribing rules for steel tube plates, however. The working pressure on the part of the tube-sheet that contains the tubes is to be found as follows:

Rule.—*Multiply 140 by the square of the plate thickness, in sixteenths of an inch, and divide the product by the square of the mean pitch of the stay-tubes, measured from center to center.*

Or,
$$P_w = \frac{140\,T^2}{P^2}$$

in which P_w = working pressure, in pounds per square inch;
T = numerator of fraction expressing plate thickness, in sixteenths of an inch;
P = mean pitch of stay-tubes.

EXAMPLE.—In a steel tube-sheet $\frac{7}{8}$ inch thick, the stay-tubes are spaced 7 inches center to center horizontally and $9\frac{3}{4}$ inches center to center vertically; what working pressure will be allowed under Canadian rules?

SOLUTION.— $\frac{7}{8} = \frac{14}{16}$, or $T = 14$.

$$P = \frac{7 + 9\frac{3}{4}}{2} = 8\frac{3}{8}$$

Applying the rule,

$$P_w = \frac{140 \times 14^2}{(8\frac{3}{8})^2} = 391.2 \text{ lb. per sq. in.} \quad \text{Ans.}$$

17. Under Canadian rules, the working pressure on that part of a steel tube-sheet that lies between the nests of tubes, if this part is not covered by a doubling plate, is to be found as follows:

Rule.—*Multiply the constant corresponding to the conditions by the square of the tube plate thickness, in sixteenths of an inch, and divide the product by the square of the center-to-center distance, in inches, of the bounding rows of tubes.*

Or, $$P_w = \frac{C\,T^2}{P^2}$$

in which P_w = working pressure, in pounds per square inch;
T = numerator of fraction expressing thickness of tube plate, in sixteenths of an inch;
P = horizontal distance from center to center of the bounding row of tubes;
C = 120 where the stay-tubes in the bounding rows are pitched with two plain tubes between them and are not fitted with nuts outside the plates;
C = 130 if the stay-tubes are fitted with nuts outside the plates and there are two plain tubes between the stay-tubes in the bounding rows;
C = 140 if each alternate tube in the bounding rows is a stay-tube and not fitted with nuts outside the plates;
C = 150 if each alternate tube in the bounding rows is a stay-tube and fitted with nuts outside the plates;
C = 160 if every tube in the bounding rows is a stay-tube but not fitted with nuts outside the plates;
C = 170 if every tube in the bounding rows is a stay-tube and each alternate stay-tube is fitted with nuts outside the plates.

EXAMPLE.—A tube-sheet is $\frac{3}{4}$ inch thick and made of steel. In the two vertical rows of tubes bounding two nests, each third tube is a

stay-tube, the stay-tubes being fitted with nuts outside the tube-sheet. The horizontal center-to-center distance of the stay-tubes in the bounding rows being 13 inches, what working pressure will be allowed, under Canadian rules, on that part of the tube-sheet situated between the nests?

SOLUTION.— $\frac{3}{4} = \frac{12}{16}$, or $T = 12$. For this case, $C = 130$. Applying the rule,

$$P_w = \frac{130 \times 12^2}{13^2} = 110.77 \text{ lb. per sq. in.} \quad \text{Ans.}$$

18. When the part of a steel tube-sheet that lies between the nests of tubes is reinforced by a doubling plate securely riveted thereto and having a thickness of at least two-thirds of the tube-sheet thickness, the Canadian rules specify that the working pressure allowable on this part shall be found as follows:

Rule.—*Multiply the constant corresponding to the case (given in Art.* **17***) by the square of the sum of the tube-sheet thickness and half the doubling-plate thickness, in sixteenths of an inch. Divide the product by the horizontal distance from center to center between the bounding rows of tubes.*

Or,
$$P_w = \frac{C\left(T + \frac{t}{2}\right)^2}{P^2}$$

in which t is the numerator of the fraction expressing the thickness of the doubling plate, in sixteenths of an inch, and the other letters have the same meaning and C the same values as in Art. **17**.

EXAMPLE.—A steel tube plate $\frac{5}{8}$ inch in thickness is reinforced by a doubling plate $\frac{1}{2}$ inch thick; the tubes in the bounding rows are 13 inches center to center horizontally, and each alternate tube is a stay-tube fitted with nuts outside the plates. What working pressure, under Canadian rules, will be allowed on the part between the nests of tubes?

SOLUTION.— $\frac{5}{8} = \frac{10}{16}$, or $T = 10$. $\frac{1}{2} = \frac{8}{16}$, or $t = 8$. C, by Art. **17**, $= 150$. Applying the rule,

$$P_w = \frac{150 \times (10 + \frac{8}{2})^2}{13^2} = 173.96 \text{ lb. per sq. in.} \quad \text{Ans.}$$

19. The tube-sheet that forms part of a firebox or combustion chamber is subjected to a crushing stress. The

magnitude of this stress is limited by the rule given in this article, which appears in the American, Board of Trade, and Canadian rules.

Rule.—*From the least horizontal distance between tube centers, in inches, subtract the inside diameter of the tubes, in inches. Multiply the remainder by the thickness of the tube plate, in inches, and the required constant. Divide the product by the product of the extreme width (measured in the direction of the length of the boiler) of the firebox or combustion chamber and the least horizontal distance between tube centers, both in inches.*

Or,
$$B = \frac{(D - d)\,TC}{WD}$$

in which B = working pressure, in pounds per square inch;
D = least horizontal distance between tube centers, in inches;
d = inside diameter of ordinary tubes, in inches;
T = tube-sheet thickness, in inches;
W = width of firebox or combustion chamber, in inches, or distance between tube-sheets in double-ended boilers in case the combustion chamber is common to the furnaces at both ends;
C = 28,000 for steel under American, Board of Trade, and Canadian rules;
C = 18,000 for iron under Canadian rules;
C = 22,000 for iron under Board of Trade rules.

EXAMPLE.—What working pressure, calculated for crushing and under American rules, will be allowed on a steel back tube-sheet of a Scotch boiler if the sheet is $\frac{3}{4}$ inch thick, the tubes are $2\frac{1}{4}$ inches in inside diameter and spaced $3\frac{1}{2}$ inches center to center horizontally, and the combustion chamber is 25 inches wide?

SOLUTION.— C = 28,000. Applying the rule,

$$B = \frac{(3\frac{1}{2} - 2\frac{1}{4}) \times \frac{3}{4} \times 28{,}000}{25 \times 3\frac{1}{2}} = 300 \text{ lb. per sq. in.} \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE

1. Under American rules, what pressure will be allowed on the inner side plates of the firebox of a firebox boiler if the plates are $\frac{1}{2}$ inch thick and stayed by ordinary riveted screw staybolts having a pitch of 6 inches? Ans. 213.3 lb. per sq. in.

2. The front head of a Scotch boiler is stayed by rods 14 inches center to center that are each fitted with a washer of proper diameter riveted to the outside of the head and one nut inside the head and one nut outside the washer. The head is $\frac{11}{16}$ inch thick; the washers are $\frac{3}{8}$ inch thick. What working pressure will be allowed, under American rules? Ans. 150.98 lb. per sq. in.

3. Under Board of Trade rules, what pressure will be allowed on the outer firebox plates, which are not exposed to heat or flame, of a firebox boiler if the plates are of steel $\frac{7}{16}$ inch thick and the screw staybolts, which are riveted over, have a pitch of 6 inches? Take C as 77. Ans. 164.27 lb. per sq. in.

4. The back head of a single-ended Scotch boiler made of steel is $\frac{3}{4}$ inch and reinforced by a doubling plate $\frac{1}{2}$ inch thick; the stayrods have nuts inside and outside the plates, and are pitched 16 inches center to center. What working pressure will be allowed, under Board of Trade rules? Take C as 150. Ans. 150 lb. per sq. in.

5. What working pressure will be allowed, under Canadian rules, on the head of example 4? Ans. 175 lb. per sq. in.

6. A flat steel plate $\frac{3}{4}$ inch thick and not exposed to the fire in any way is supported by stayrods having a pitch of 13 inches; the stayrods are fitted with nuts and washers, the washers being $\frac{1}{2}$ inch thick and $4\frac{1}{2}$ inches in diameter. What working pressure will be allowed on this plate, under Canadian rules? Ans. 159.76 lb. per sq. in.

7. A double-ended Scotch boiler has a single combustion chamber common to all furnaces. The combustion-chamber tube-sheets contain three nests of tubes and are $\frac{3}{4}$ inch thick, and made of steel. The stay-tubes in the bounding rows of tubes are pitched $9\frac{1}{2}$ inches vertically and 12 inches horizontally, and are fitted with nuts. What working pressure, under Board of Trade rules, will be allowed on the parts of the tube plates that are situated between the nests of tubes? Take C as 100. Ans. 152.1 lb. per sq. in.

8. What pressure will be allowed on the tube-sheets in example 7, under Canadian rules? There are two plain tubes between each two stay-tubes. Ans. 130 lb. per sq. in.

9. A steel tube plate $\frac{1}{2}$ inch thick is reinforced by a doubling plate $\frac{3}{8}$ inch thick riveted to the part between the nests of tubes. In

the bounding rows, each second tube is a stay-tube fitted with nuts; the horizontal center-to-center distance of the stay-tubes in the bounding rows is 11 inches. Under Canadian rules, what working pressure will be allowed on the reinforced part of the tube-sheet?

Ans. 150 lb. per sq. in.

10. What pressure, calculated for crushing and under Board of Trade rules, may be carried on an iron back tube-sheet $\frac{1}{2}$ inch thick containing tubes having an inside diameter of $2\frac{1}{2}$ inches and spaced 4 inches center to center, the combustion chamber being 22 inches wide?

Ans. 187.5 lb. per sq. in.

STRENGTH OF STAYS

20. The area supported by one stay is found by multiplying the distance from center to center of stays in one direction by the distance from center to center in the other

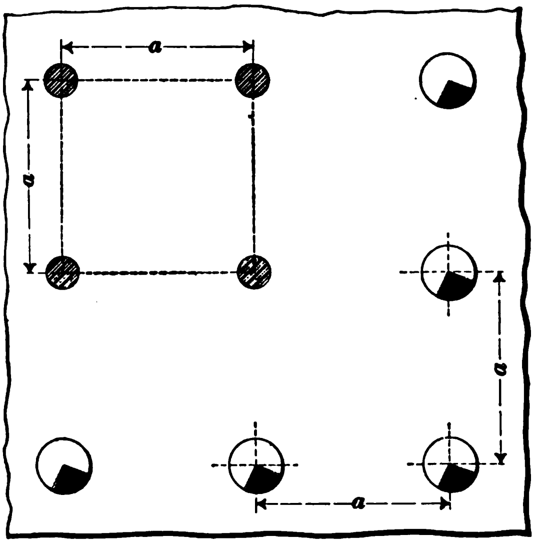


FIG. 1

direction. The center-to-center distance is known as the *pitch* of the stays. In Fig. 1, the stays being spaced equidistant in both directions, the area supported by each stay is $a \times a = a^2$. If the pitch is 7 inches and 6 inches, respectively, the area supported by each stay will be $7 \times 6 = 42$ square inches. If extreme accuracy is required, the

area of the staybolt should be subtracted from the area found by multiplying the pitch in one direction by the pitch in the other direction. Thus, in the example, assuming the area of the staybolt to be 1 square inch, the area supported by the bolt will be $42 - 1 = 41$ square inches. This is a refinement of calculation rarely used in practice, but it is the mathematically correct way of calculating the area. Now, the area supported by each stay, in square inches, multiplied by the *gauge pressure* in pounds, constitutes the load borne by each stay. To bear the load safely, the stay must have a certain minimum area, which depends on the maximum stress allowable per square inch of cross-section.

By "area of a stay" is always meant the smallest cross-sectional area of the stay. In the case of a screw stay, the smallest cross-sectional area is that corresponding to the diameter over the bottom of the thread, unless some other part has a smaller diameter.

21. Under American rules, the maximum stress allowable on stays is as follows: Tested steel stays above $2\frac{1}{2}$ inches, 9,000 pounds per square inch; tested steel stays $1\frac{1}{4}$ inches and not exceeding $2\frac{1}{2}$ inches, when such stays have neither been forged nor welded, excepting upsetting of the ends and subsequent annealing, 8,000 pounds per square inch; tested Huston or similar type of brace having a cross-sectional area exceeding 5 square inches, 8,000 pounds per square inch; tested Huston or similar type of brace having a cross-sectional area of not less than 1.227 and more than 5 square inches, provided the braces are prepared at one heat from a solid piece of plate without welds, 7,000 pounds per square inch; all stays not otherwise provided for, 6,000 pounds per square inch.

22. The Board of Trade rules, in case of new boilers, fix the maximum stress on stays as follows: Solid iron screw stays, 7,000 pounds per square inch when the stays have not been welded; if iron stays have been welded, 5,000 pounds per square inch; iron stay-tubes having a net thickness of at least $\frac{1}{4}$ inch, 6,000 pounds per square inch;

solid steel screw stays, tested and not welded, 9,000 pounds per square inch; tested steel stay-tubes having a net thickness of at least $\frac{1}{4}$ inch, 7,500 pounds per square inch; stays for convex heads thick enough for the pressure not to require staying, 14,000 pounds per square inch when not welded and 10,000 pounds per square inch when welded.

23. The Canadian rules, in case of new boilers, specify the maximum stress on stays as follows: Solid iron screw stays, unwelded or worked in the fire, 7,000 pounds per square inch; solid iron screw stays which have been welded or worked in the fire, 6,000 pounds per square inch; tested solid steel screw stays, unwelded, 9,000 pounds per square inch; tested steel stayrods exceeding $1\frac{1}{2}$ inches in diameter, 10,000 pounds per square inch; tested steel stay-tubes having a net thickness of at least $\frac{1}{4}$ inch, 7,500 pounds per square inch.

For water-tube boilers, the allowable stress on solid iron screw stays, not welded, when supporting flat surfaces, must not exceed 5,000 pounds per square inch; when such stays have been welded the stress must not exceed 4,000 pounds per square inch. When stays are used under conditions similar to those in fire-tube boilers, the stress may be 6,000 pounds per square inch. When stays are used for supporting heads of steam drums, the heads being bumped and heavy enough to pass without staying, a stress of 10,000 pounds per square inch is allowed.

24. The rules given in this article and relating to stays and staying should be very carefully studied and memorized, especially by candidates for an American marine engineer's license, as the Examining Board is enjoined by law to reject candidates who cannot solve problems similar to those given in connection with the various rules.

Let A = area, in square inches, supported by a screw stay, staybolt, or stayrod;

B = cross-sectional area of stay, in square inches;

P = working steam pressure, in pounds per square inch;

L = load on stay, in pounds;
S = stress per square inch of cross-section of stay;
S_t = total allowable stress in a stay;
M = lawful stress per square inch of cross-section of stay;
A_t = total area to be supported, in square inches;
N = number of stays.

Rule I.—*To find the load on a stay, multiply the area supported by the stay, in square inches, by the gauge pressure of the steam, in pounds per square inch.*

Or, $$L = AP \qquad (1)$$

EXAMPLE 1.—Find the load on a staybolt, the pitch of the staybolts being 8 inches either way and the steam pressure 85 pounds.

SOLUTION.—Applying the rule,
$$L = 8^2 \times 85 = 5{,}440 \text{ lb. Ans.}$$

Rule II.—*To find the stress per square inch of cross-section in a stay, divide the load on the stay, in pounds, by the area of the stay, in square inches.*

Or, $$S = \frac{AP}{B} \qquad (2)$$

EXAMPLE 2.—Staybolts $1\frac{1}{4}$ inches in diameter are placed $8\frac{1}{2}$ inches from center to center; the steam pressure being 100 pounds, what is the stress per square inch in the stay?

SOLUTION.—The area of the staybolt is $(1\frac{1}{4})^2 \times .7854 = 1.2272$ sq. in. Applying rule II,
$$S = \frac{(8\frac{1}{2})^2 \times 100}{1.2272} = 5{,}887.39 \text{ lb. Ans.}$$

Rule III.—*To find the total allowable stress in a stay, multiply the area of the stay, in square inches, by the lawful stress per square inch.*

Or, $$S_t = BM \qquad (3)$$

EXAMPLE 3.—Under Canadian rules, what stress can be allowed on a steel stayrod $2\frac{1}{2}$ inches in diameter?

SOLUTION.—By Art. **23**, $M = 10{,}000$. Applying the rule,
$$S_t = (2\frac{1}{2})^2 \times .7854 \times 10{,}000 = 49{,}087 \text{ lb. Ans.}$$

Rule IV.—*To find the working steam pressure allowable on a stay, in pounds per square inch, multiply the area of the stay,*

in square inches, by the stress corresponding to the diameter and material of the stay, in pounds per square inch. Divide this product by the area supported by the stay, in square inches.

Or, $$P = \frac{BM}{A} \quad (4)$$

EXAMPLE 4.—Steel stayrods 2 inches in diameter are placed 14 inches from center to center; what working steam pressure would be allowed for these stays, under American rules?

SOLUTION.—By Art. **21**, M = 8,000. Applying the rule,

$$P = \frac{2^2 \times .7854 \times 8{,}000}{14^2} = 128.23 \text{ lb. Ans.}$$

Rule V.—*To find the number of stays required, multiply the area of the sheet to be stayed, in square inches, by the steam pressure, in pounds per square inch, and divide by the total stress allowable in the given size of stay, in pounds.*

Or, $$N = \frac{A_t P}{S_t} \quad (5)$$

EXAMPLE 5.—Under Canadian rules, how many $1\frac{1}{8}$-inch round iron staybolts are required for a plate 40 inches by 50 inches to carry 90 pounds per square inch working pressure? The staybolts have been worked in the fire.

SOLUTION.—By Art. **23**, M = 6,000. Then, by rule III,

$$S_t = (1\tfrac{1}{8})^2 \times .7854 \times 6{,}000 = 5{,}964 \text{ lb., nearly}$$

Applying rule V,

$$N = \frac{40 \times 50 \times 90}{5{,}964} = 30.2, \text{ say 31 stays. Ans.}$$

Rule VI.—*To find the area of a direct stay, in square inches, multiply the area supported by the stay, in square inches, by the steam pressure, in pounds per square inch. Divide the product by the lawful stress per square inch of cross-section.*

Or, $$B = \frac{AP}{M} \quad (6)$$

EXAMPLE 6.—Wrought-iron screw stays that have not been welded are pitched 7 inches center to center; the working pressure desired being 180 pounds per square inch, what should the area of each stay be, under Board of Trade rules?

SOLUTION.—By Art. **22**, M = 7,000. Applying rule VI,

$$B = \frac{7^2 \times 180}{7{,}000} = 1.26 \text{ sq. in. Ans.}$$

25. While the load on all direct stays is equal to the total pressure on the area supported by the stay, it is greater for palm stays.

Let b = length, in inches, of a line drawn at right angles to surface supported to end of diagonal stay, that is, distance, in inches, from boiler head to intersection of center line of stay with shell, in the usual case of head being at right angles to shell;

l = length of palm stay, in inches, measured between head and shell along center line of stay;

AP = total pressure, in pounds, on area supported by stay;

L = load on stay, in pounds.

Rule I.—*To find the load, in pounds, on a palm stay, divide the length of the palm stay, in inches, by the distance from the head to the intersection of the center line of the palm stay with the shell, in inches. Multiply the quotient by the total pressure, in pounds.*

Or,
$$L = \frac{l}{b} A P \qquad (1)$$

EXAMPLE 1.—The palm stay of a boiler is 78 inches long; the distance from the head to the point of intersection of the center line of the stay with the shell is 72 inches. If the area supported by the stay is 100 square inches and the steam pressure 80 pounds, what is the load on the stay?

SOLUTION.—The total steam pressure is $100 \times 80 = 8{,}000$ lb. Then, applying the rule,

$$L = \frac{78}{72} \times 8{,}000 = 8{,}666.67 \text{ lb. Ans.}$$

The stress per square inch of cross-section is found for a palm stay in the same manner as for any other stay, that is, by rule II, Art. **24**.

EXAMPLE 2.—What is the stress per square inch if the stay in the previous example is $1\frac{5}{8}$ inches in diameter?

SOLUTION.—Applying rule II, Art. **24**,

$$S = \frac{8{,}666.67}{(1\frac{5}{8})^2 \times .7854} = 4{,}178.92 \text{ lb. Ans.}$$

The load for a given supported area of plate and steam pressure being greater for palm stays than for stayrods, the allowable working pressure is conversely smaller.

Rule II.—*To find the working steam pressure allowable on a palm stay, multiply the result obtained by rule IV, Art. 24, by the quotient obtained by dividing the distance from the head to the intersection of the center line of the stay with the shell by the length, both in inches.*

Or, $$P = \frac{B\,M}{A} \times \frac{b}{l} \qquad (2)$$

in which B, M, A, and P have the same meaning as in Art. 24.

EXAMPLE 3.—What working steam pressure will be allowed for the palm stay having the same dimensions as given in the last example, made of iron, and allowed a stress of 6,000 pounds per square inch?

SOLUTION.—Applying rule II,

$$P = \frac{(1\tfrac{5}{8})^2 \times .7854 \times 6{,}000}{100} \times \frac{72}{78} = 114.86 \text{ lb. Ans.}$$

Rule III.—*To find the area of a palm stay, find the area of a direct stay required to support the surface (by rule VI, Art. 24); multiply this area by the length of the palm stay, in inches, and divide the product by the length of a line drawn at right angles to the surface supported to the end of the palm stay.*

Or, $$B = \frac{A\,P}{M} \times \frac{l}{b} \qquad (3)$$

in which B, A, P, and M have the same meaning as in Art. 24.

EXAMPLE 4.—Find the diameter of a wrought-iron palm stay, with the foot welded on, which supports an area of 140 square inches and is to be strong enough to stand a working pressure of 160 pounds per square inch; the length of the stay is 66 inches, and the length of a line at right angles to the boiler head to the end of the stay is 60 inches. Take the permissible stress per square inch as 5,000 pounds, as prescribed by the Board of Trade rules.

SOLUTION.—Applying the rule,

$$B = \frac{140 \times 160}{5{,}000} \times \frac{66}{60} = 4.928 \text{ sq. in.}$$

Then, diameter of stay is $\sqrt{\frac{4.928}{.7854}} = 2.505$ in. In practice, the next

larger commercial size of round iron would be used. This is either $2\frac{9}{16}$ or $2\frac{5}{8}$ in., depending on the practice of the mill from which the iron is purchased. Ans.

26. The American, Board of Trade, and Canadian rules all give the same rule for the working pressure allowable on solid rectangular girder stays, sometimes called *crown bars*, and used for supporting the top sheet of combustion chambers and fireboxes, differing only in the value of the constant employed. The American rules provide that when the pressure exceeds 160 pounds per square inch, girder stays must be suspended from the top of the shell by braces having each a sectional area at least twice the sectional area of each of the staybolts suspending the top sheet from the girder.

Rule.—*Multiply the constant corresponding to conditions by the square of the depth of the girder, in inches, and the thickness of the girder, in inches. Divide this product by the difference between the width of the combustion box, in inches (measured in the direction the girder is applied) and the pitch of the staybolts, multiplied by the distance between girders from center to center, in inches, and the length of the girder, in feet.*

Or,
$$B = \frac{C d^2 T}{(W - P) D L}$$

in which B = pressure, in pounds per square inch;
d = depth of girder, in inches;
T = thickness of girder, in inches;
W = width of combustion box, in inches;
P = pitch of staybolts, in inches;
D = distance between girders, in inches, measured from center to center;
L = length of girder, in feet;
N = number of supporting bolts;
C = 550, American rules, when the girder is fitted with one staybolt;
C = 825, American rules, when the girder is fitted with two or three staybolts;
C = 935, American rules, when the girder is fitted with four staybolts;

$C = \frac{N\,1{,}200}{N+1}$, Board of Trade rules, when the number of staybolts is odd and the girder stays are iron;

$C = \frac{N\,1{,}200}{N+1} \times 1.1$, Board of Trade rules, when the number of staybolts is odd and the girder stays are steel;

$C = \frac{(N+1)\,1{,}200}{N+2}$, Board of Trade rules, when the number of staybolts is even and the girder stays are iron;

$C = \frac{(N+1)\,1{,}200}{N+2} \times 1.1$, Board of Trade rules, when the number of staybolts is even and the girder stays are steel;

$C = \frac{N\,1{,}000}{N+1}$, Canadian rules, when the number of staybolts is odd and the girder stays are iron;

$C = \frac{N\,1{,}000}{N+1} \times 1.1$, Canadian rules, when the number of staybolts is odd and the girder stays are steel;

$C = \frac{(N+1)\,1{,}000}{N+2}$, Canadian rules, when the number of staybolts is even and the girder stays are iron;

$C = \frac{(N+1)\,1{,}000}{N+2} \times 1.1$, Canadian rules, when the number of staybolts is even and the girder stays are steel.

EXAMPLE 1.—Under American rules, what working pressure will be allowed on a solid rectangular steel girder 7 inches deep and 2 inches thick, fitted with three staybolts, and 2.5 feet long? The girders are spaced 7 inches center to center, the pitch of the staybolts is 7.5 inches, and the combustion chamber is 30 inches wide.

SOLUTION.—For this case, C = 825. Applying the rule,

$$B = \frac{825 \times 7^2 \times 2}{(30 - 7.5) \times 7 \times 2.5} = 205.3 \text{ lb. per sq. in. Ans}$$

EXAMPLE 2.—What working pressure will be allowed under Board of Trade rules on the girder in example 1?

SOLUTION.—For this case,

$$C = \frac{3 \times 1{,}200}{3 + 1} \times 1.1 = 990$$

Applying the rule,

$$B = \frac{990 \times 7^2 \times 2}{(30 - 7.5) \times 7 \times 2.5} = 246.4 \text{ lb. per sq. in. Ans.}$$

EXAMPLE 3.—What working pressure will be allowed under Canadian rules on the girder in example 1?

SOLUTION.—For this case,

$$C = \frac{3 \times 1{,}000}{3 + 1} \times 1.1 = 825$$

Applying the rule,

$$B = \frac{825 \times 7^2 \times 2}{(30 - 7.5) \times 7 \times 2.5} = 205.3 \text{ lb. per sq. in. Ans.}$$

EXAMPLES FOR PRACTICE

1. Find the load on a staybolt, the pitch being $6\frac{1}{2}$ inches and the steam pressure 125 pounds per square inch. Ans. 5,281.25 lb.

2. Find the stress per square inch of section of a staybolt 1 inch in diameter, the pitch being $5\frac{1}{2}$ inches and the steam pressure 120 pounds. Ans. 4,621.85 lb. per sq. in.

3. Under Board of Trade rules, what stress can be allowed on an iron stayrod 2 inches in diameter and not welded? Ans. 21,991.2 lb.

4. Under American rules, what working pressure will be allowed on $1\frac{1}{4}$-inch steel staybolts pitched $8\frac{1}{2}$ inches center to center? The staybolts have not been forged or welded. Ans. 135.88 lb. per sq. in.

5. The top sheet of the middle combustion chamber of a Scotch double-ended boiler is 34 inches by 24 inches and is stayed by $1\frac{1}{4}$-inch steel staybolts; how many solid steel staybolts will be required for 160 pounds per square inch working pressure, under Board of Trade rules? Ans. 12

6. Wrought-iron screw stays pitched 8 inches center to center support the plates of a firebox; what should be the area of each for a working pressure of 160 pounds per square inch, under American rules? Ans. 1.707 sq. in.

7. What is the stress per square inch of section in a palm stay 7 feet 6 inches long? The diameter of the stay is 2 inches, the area supported by the stay is 156 square inches, and the steam pressure is

120 pounds. The distance from the head to the point of intersection of the center line of the stay with the shell is 6 feet 9 inches.

Ans. 6,620.83 lb.

8. What working pressure, under Canadian rules, will be allowed on a welded wrought-iron palm stay $2\frac{1}{4}$ inches in diameter, if the stay is 76 inches long and supports a surface of 140 square inches; the length of a line at right angles to the boiler head and to the end of the stay is 68 inches? Ans. 152.47 lb. per sq. in.

9. Find the area of a wrought-iron palm stay, with the foot welded on, that is to support an area of 164 square inches against a steam pressure of 180 pounds per square inch. The palm stay is 62 inches long, and the length of a line at right angles to the boiler head to the end of the stay is 57 inches. Inspection is under Board of Trade rules.

Ans. 6.422 sq. in.

10. Under American rules, what working pressure will be permitted on a girder stay fitted with four staybolts having a pitch of 7 inches, the girder stay being $2\frac{1}{2}$ inches thick, having a depth of 8 inches, and a length of 2.9 feet? The girder stays are placed 7 inches center to center and the combustion chamber is 35 inches wide.

Ans. 263.19 lb. per sq. in.

FURNACE FLUES, SMOKE FLUES, AND TUBES

FURNACE FLUES

27. The American, Board of Trade, and Canadian rules prescribe the same general rule for finding the permissible external working pressure on corrugated and similar furnace flues, differing somewhat, however, in the value of the constant forming a factor in the rule.

Rule.—*Multiply the constant corresponding to the case by the thickness of the furnace flue, in inches, and divide the product by the mean diameter, in inches.*

Or,
$$P = \frac{CT}{D}$$

in which P = working pressure, in pounds per square inch;
T = thickness of furnace, in inches;
D = mean diameter, in inches;
C = 10,000 for corrugated iron furnace flues, Canadian rules;

C = 15,000 for bulb-type furnace flues made by the Leeds Forge Company, Board of Trade rules;

C = 15,600 for Morison furnace flues, American rules;

C = 14,000 for Morison furnace flues, under Board of Trade and Canadian rules, and for Purves, Fox, and Brown furnace flues under American, Board of Trade, and Canadian rules.

Under American rules, the mean diameter of Morison furnace flues is the least inside diameter plus 2 inches; under Board of Trade and Canadian rules, it is the outside diameter at the bottom of the corrugations. For ordinary corrugated (Fox) furnace flues, the mean diameter, under all rules, is the outside diameter at the bottom of the corrugations. For Purves and Brown furnace flues, under all rules, the mean diameter is the least outside diameter, that is, the diameter of the plain parts. For Leeds Forge Company furnace flues, under Board of Trade rules, the mean diameter is the outside diameter at the middle of the plain parts.

EXAMPLE 1.—What working pressure will be allowed, under Canadian rules, on an iron corrugated furnace flue 40 inches in mean diameter and $\frac{1}{2}$ inch thick?

SOLUTION.— C = 10,000. Applying the rule,

$$P = \frac{10{,}000 \times \frac{1}{2}}{40} = 125 \text{ lb. per sq. in.} \quad \text{Ans.}$$

EXAMPLE 2.—What working pressure will be allowed on a Purves furnace flue 36 inches in mean diameter and $\frac{3}{8}$ inch thick?

SOLUTION.— C = 14,000. Applying the rule,

$$P = \frac{14{,}000 \times \frac{3}{8}}{36} = 145.83 \text{ lb. per sq. in.} \quad \text{Ans.}$$

28. A type of furnace flue is used to some extent which is built up of short sections fitted into each other and riveted together, each section having one corrugation $2\frac{1}{2}$ inches deep, and the corrugations being 18 inches center to center. The plain parts at the end do not exceed 12 inches in length, and

the thickness is not less than $\frac{7}{16}$ inch. For such a furnace, the American rules prescribe that the working pressure be found by the rule in Art. **27**, making $C = 10{,}000$.

29. Plain horizontal furnace flues made up of flanged sections riveted together with a ring between the flanges are known as the Adamson type of furnace flue. Under the latest American rules, the sections must not be shorter than 23 inches nor longer than 54 inches, nor less than $\frac{5}{16}$ inch thick. The flanges must have a depth of not less than three times the rivet-hole diameter plus the radius of the furnace wall (measured from the inside of the furnace flue); the radii of the flanges on the fire-side should be not less than three times the plate thickness. The distance from the edge of the rivet hole to the edge of the flange must not be less than the diameter of the rivet hole; the rivet diameter before riveting must be at least $\frac{1}{4}$ inch more than the plate thickness. The rings must be at least $\frac{1}{2}$ inch thick and have a depth of not less than three times the rivet-hole diameter; the fire-edge of the ring must terminate at or about the point of tangency to the curve of the flange.

Rule.—*Divide 51.5 by the outside diameter of the furnace flue, in inches. Multiply the quotient by the difference between 18.75 times the plate thickness, in sixteenths of an inch, and the product of the length of the furnace flue section, in inches, and 1.03.*

Or,
$$P = \frac{51.5}{D}\,[18.75\,T - (L \times 1.03)]$$

in which P = working pressure, in pounds per square inch;
D = outside diameter of furnace flue, in inches;
T = numerator of fraction expressing plate thickness, in sixteenths of an inch;
L = length of furnace section, in inches.

EXAMPLE.—An Adamson type of furnace flue is made of sections 24 inches long and $\frac{7}{16}$ inch thick; the diameter being 40 inches, what working pressure will be allowed under American rules?

SOLUTION.— $T = 7$. Applying the rule,

$$P = \frac{51.5}{40} \times [18.75 \times 7 - (24 \times 1.03)] = 137.16 \text{ lb. per sq. in.} \quad \text{Ans.}$$

30. For horizontal steel furnace flues of the Adamson type, the Board of Trade and Canadian rules prescribe the following rule for finding the working pressure:

Rule.—*Multiply 9,900 by the plate thickness, in inches, and divide the product by three times the outside diameter of the furnace flue, in inches. Multiply the quotient by the difference between 5 and the quotient obtained by dividing the sum of the length between centers of flanges, in inches, and 12 by sixty times the plate thickness, in inches.*

Or,
$$B = \frac{9{,}900\,T}{3\,D}\left(5 - \frac{L+12}{60\,T}\right)$$

in which B = working pressure, in pounds per square inch;
T = plate thickness, in inches;
D = outside diameter of furnace flue, in inches;
L = length of section between center of flanges, in inches.

The rules provide that L must be never greater than $120\,T - 12$.

EXAMPLE.—What working pressure will be allowed, under Board of Trade and Canadian rules, on the furnace flue mentioned in the example in Art. **29**?

SOLUTION.—Applying the rule,

$$B = \frac{9{,}900 \times \frac{7}{16}}{3 \times 40} \times \left(5 - \frac{24+12}{60 \times \frac{7}{16}}\right) = 130.97 \text{ lb. per sq. in.} \quad \text{Ans.}$$

31. The working pressure allowable on plain cylindrical furnace flues, under American rules, is found by the rule given in Art. **29**, taking L as the center-to-center distance of the strengthening rings.

32. The Board of Trade and Canadian rules prescribe the same rules for finding the working pressure allowable on plain cylindrical furnace flues, taking as a standard a furnace with longitudinal joints welded, or made with a single butt strap double riveted, or double butt straps single riveted, rivet holes drilled. Such furnaces are allowed the highest working pressure, determined in part by the constant 90,000 in rule I, this article; for furnaces constructed in a different manner than just specified, the constant is reduced. The

working pressure is to be calculated both by rule I and by rule II; it will be the smaller of the two values. The rules given apply to iron furnace flues; for steel furnace flues multiply the results of the two rules by 1.1 under Board of Trade rules; under Canadian rules multiply the result of rule I by 1.1, and in rule II substitute the constant 10,000 for 9,000 in case the furnace flue is made of steel.

Rule I.—*Multiply the constant 90,000 or such other constant as the case demands by the square of the plate thickness, in inches. Divide the product by the product obtained by multiplying the sum of the length of the furnace flue, in feet, and 1 by the diameter, in inches.*

Or,
$$B = \frac{90{,}000\, T^2}{(L+1)\, D} \qquad (1)$$

Rule II.—*Multiply 9,000 by the plate thickness, in inches, and divide the product by the diameter, in inches.*

Or,
$$B = \frac{9{,}000\, T}{D} \qquad (2)$$

In the rules given, let

B = working pressure, in pounds per square inch;
T = plate thickness, in inches;
L = length, in feet, measured from center to center of strengthening rings if such are fitted;
D = outside diameter of furnace, in inches.

The constants to be used in place of 90,000, are as follows, the joints referred to being the longitudinal joints:

80,000 for single-butt-strap joints single riveted, drilled rivet holes;
85,000 for single-butt-strap joints double riveted, punched rivet holes;
75,000 for single-butt-strap joints single riveted, punched rivet holes;
85,000 for double-butt-strap joints single riveted, punched rivet holes;
80,000 for double-riveted lap joints, beveled, drilled rivet holes;

75,000 for double-riveted lap joints, not beveled, drilled rivet holes;

70,000 for single-riveted lap joints, beveled, drilled rivet holes;

65,000 for single-riveted lap joints, not beveled, drilled rivet holes;

75,000 for double-riveted lap joints, beveled, punched rivet holes;

70,000 for double-riveted lap joints, not beveled, punched rivet holes;

65,000 for single-riveted lap joints, beveled, punched rivet holes;

60,000 for single-riveted lap joints, not beveled, punched rivet holes.

EXAMPLE 1.—What working pressure will be allowed on a plain cylindrical furnace flue 42 inches in diameter, $\frac{1}{2}$ inch thick, made of steel with welded longitudinal joint, fitted with strengthening rings 2 feet apart, under Board of Trade rules?

SOLUTION.—In this case, the constant is 90,000. Applying rule I, and multiplying by 1.1 on account of the furnace flue being steel,

$$B = \frac{90{,}000 \times (\frac{1}{2})^2}{(2+1) \times 42} \times 1.1 = 196.43 \text{ lb. per sq. in}$$

Applying rule II and multiplying by 1.1

$$B = \frac{9{,}000 \times \frac{1}{2}}{42} \times 1.1 = 117.86 \text{ lb. per sq. in.}$$

The result of rule II being the smaller, it is the working pressure. Ans.

EXAMPLE 2.—Under Canadian rules, what working pressure will be allowed on a plain cylindrical steel furnace flue 40 inches in diameter having the longitudinal seam fitted with a single-riveted single butt strap, drilled rivet holes, plate $\frac{1}{2}$ inch thick, and strengthening rings placed $3\frac{1}{2}$ feet center to center?

SOLUTION.—For this case, the constant is 80,000. Applying rule I and multiplying by 1.1,

$$B = \frac{80{,}000 \times (\frac{1}{2})^2}{(3\frac{1}{2}+1) \times 40} \times 1.1 = 122.22 \text{ lb. per sq. in.}$$

Under Canadian rules, the constant 9,000 in rule II becomes 10,000 for steel. Applying the rule,

$$B = \frac{10{,}000 \times \frac{1}{2}}{40} = 125 \text{ lb. per sq. in.}$$

The result of rule I being the smaller, it is the working pressure in this case. Ans.

33. When plain cylindrical flues are used for furnaces of vertical fire-tube boilers, the working pressure allowable on them is, under American rules, found by the rule in Art. **27**, making $C = 10{,}577$. The length of the furnace flue must not exceed 42 inches, however, measuring from the center of the rivet holes in the head to the center of the rivet holes in the leg. If the diameter of the furnace flue for a vertical boiler exceeds 42 inches, it is considered to be a flat surface and must be braced by stays.

34. Under Board of Trade and Canadian rules, the working pressure allowable on plain cylindrical furnace flues used as furnaces for vertical fire-tube boilers is to be found by applying the two rules given in Art. **32**, reducing the constant corresponding to the construction 10 per cent., that is, multiplying it by .9.

35. Vertical fire-tube boilers are often fitted with a cone-shaped combustion chamber at the top. Under American rules, the working pressure is found by the rule in Art. **27**, making $C = 10{,}153$. D is taken as the outside diameter at the middle of the height, and must not exceed 42 inches. When larger, the cone must be stayed as a flat surface. The height must not exceed 42 inches, measured from the center of the rivet holes in the top head to the center of the rivet holes in the upper tube-sheet.

SMOKE FLUES

36. Under American rules, the Morison, Fox, Brown, Purves, and Adamson types of furnace flues may be used as smoke flues for steam chimneys (superheaters). These types are almost invariably made of steel, and when used as smoke flues of superheaters their working pressure is ascertained by the rule in Art. **27**, making $C = 12{,}000$, provided the conditions here given are complied with.

Flues under 30 inches in diameter must be at least $\frac{5}{16}$ inch thick and be supported by angle rings at least $2\frac{1}{2}$ inches by $2\frac{1}{2}$ inches.

Flues 30 inches and under 45 inches in diameter must be at least $\frac{3}{8}$ inch thick and be supported by angle rings at least $2\frac{1}{2}$ inches by $2\frac{1}{2}$ inches.

Flues 45 inches and under 55 inches in diameter must be at least $\frac{7}{16}$ inch thick and be supported by angle rings at least 3 inches by 3 inches.

Flues 55 inches and under 65 inches in diameter must be at least $\frac{1}{2}$ inch thick and be supported by angle rings at least 3 inches by 3 inches.

Flues 65 inches and under 75 inches in diameter must be at least $\frac{9}{16}$ inch thick and be supported by angle rings at least $3\frac{1}{2}$ inches by $3\frac{1}{2}$ inches.

Flues 75 inches and under 85 inches in diameter must be at least $\frac{5}{8}$ inch thick and be supported by angle rings at least $3\frac{1}{2}$ inches by $3\frac{1}{2}$ inches.

Flues 85 inches in diameter must be at least $\frac{11}{16}$ inch thick and be supported by angle rings at least 4 inches by 4 inches.

For flues over 85 inches in diameter, add $\frac{1}{16}$ inch to $\frac{11}{16}$ inch for every 10 inches increase in the diameter.

The center-to-center distance between the angle rings, or the distance from the center of an angle ring to the center of the rivets in the heads must never exceed $2\frac{1}{2}$ feet. The angle rings must be accurately fitted to the flues and substantially riveted thereto, and must be connected to the outer shell by braces not more than 20 inches center to center on the flue.

When the flues are made in sections not exceeding $2\frac{1}{2}$ feet in length and united by Adamson flanged joints, the bracing may be dispensed with.

Plain cylindrical flues may be used for superheaters if their length does not exceed 8 feet, and they have a minimum thickness of $\frac{5}{8}$ inch when under 32 inches in diameter and $\frac{11}{16}$ inch when over 32 inches and under 46 inches in diameter. When these conditions are complied with, the constant C in the rule in Art. **27** is 8,000.

37. The Board of Trade and Canadian rules discourage the use of steel for superheaters. Both provide that when

iron furnace flues are used as smoke flues for superheaters, the constants given in Art. **32** should be multiplied by $\frac{30}{47}$ before applying the two rules in Art. **32**; the smaller of the two pressures will then be the allowable working pressure.

38. Under American rules, the working pressure on ordinary riveted or lap-welded smoke flues made in sections, as used in externally fired flue boilers, is found by the rule in Art. **27**, the constant C for this case being 8,000. The following conditions must be complied with: Greatest length of section for flues over 6 and not over 10 inches in diameter, 60 inches; for flues over 10 and not over 23 inches, 36 inches; for flues over 23 and not over 40 inches, 30 inches. Minimum thickness to be as follows: Flues over 6 and not over 7 inches, .18 inch; flues over 7 and not over 8 inches, .2 inch; flues over 8 and not over 10 inches, .21 inch; flues over 10 and not over 12 inches, .22 inch; for every inch increase in diameter over 12 inches, the minimum thickness is to be increased .01 inch over .22 inch.

EXAMPLE.—Find the minimum thickness of a smoke flue 24 inches in diameter, made of sections properly riveted, and find the working pressure allowed thereon.

SOLUTION.—Minimum thickness is

$$.22 + [(24 - 12) \times .01] = .34 \text{ inch. Ans.}$$

Applying the rule in Art. **27**, and making $C = 8{,}000$,

$$P = \frac{8{,}000 \times .34}{24} = 113.33 \text{ lb. per sq. in. Ans.}$$

39. Formerly, lap-welded flues over 6 inches and not over 16 inches in diameter could be made as long as 18 feet and were then allowed an external working pressure of 60 pounds per square inch under American rules; when made in sections not over 5 feet long, these flues were allowed a working pressure of 120 pounds per square inch. This has been changed, however, and in all marine boilers made in the United States of America after June 30, 1905, flues over 6 inches in external diameter must conform to the conditions stated in Art. **38**.

BOILER TUBES

40. Under American rules, lap-welded boiler tubes from 1 inch in diameter up to and including 6 inches in diameter and of standard thickness for the diameter, may be of any length and be allowed an external working pressure up to and including 225 pounds per square inch, if they are deemed safe by the inspectors. With each shipment of tubes, the manufacturers must furnish an affidavit certifying that the tubes have been properly tested.

Neither the Board of Trade nor the Canadian rules contain any clause fixing the external working pressure to be allowed on boiler tubes.

EXAMPLES FOR PRACTICE

1. What working pressure will be allowed, under American rules, on a Morison furnace flue 38 inches in mean diameter and $\frac{1}{2}$ inch thick? Ans. 205.26 lb. per sq. in.

2. A furnace flue 36 inches in mean diameter is built up from sections 18 inches long, each section having one corrugation $2\frac{1}{2}$ inches deep. The plate being $\frac{1}{2}$ inch thick, what working pressure will be allowed, under American rules? Ans. 138.89 lb. per sq. in.

3. An Adamson type furnace flue is made in sections 30 inches long and $\frac{9}{16}$ inch thick; the outside diameter being 42 inches, what working pressure will be allowed, under American rules? Ans. 169.03 lb. per sq. in.

4. What working pressure will be allowed, under Board of Trade rules, on the furnace flue in example 3? Ans. 165.98 lb. per sq. in.

5. What working pressure will be allowed, under Canadian rules, on a plain circular furnace flue made of iron $\frac{1}{2}$ inch thick having a single-riveted, single-butt-strap longitudinal joint, an outside diameter of 38 inches, and a length of 4 feet? The rivet holes are drilled. Ans. 105.26 lb. per sq. in.

6. Under American rules, what working pressure will be allowed on a plain cylindrical flue used for the furnace of a vertical boiler if its diameter is 36 inches and its thickness $\frac{3}{8}$ inch? Ans. 110.18 lb. per sq. in.

7. A plain cylindrical furnace flue $2\frac{1}{2}$ feet long, with a welded longitudinal joint, made of iron $\frac{7}{16}$ inch thick, has an outside diameter of 30 inches and is used for the furnace of a vertical boiler; what

working pressure will be allowed on it under Board of Trade rules?
Ans. 118.13 lb. per sq. in.

8. The cone-shaped upper combustion chamber of a vertical boiler has a mean diameter of 36 inches and is $\frac{3}{8}$ inch thick; what working pressure will be allowed on it, under American rules?
Ans. 105.76 lb. per sq. in.

9. A plain cylindrical furnace flue 30 inches in diameter and $\frac{5}{8}$ inch thick is used as a smoke flue for a superheater; what working pressure will be allowed on it, under American rules?
Ans. 166.67 lb. per sq. in.

10. A plain cylindrical iron furnace flue 32 inches in diameter, $\frac{5}{8}$ inch thick, and 48 inches long, with a welded longitudinal joint, is used as a smoke flue for a superheater; what working pressure will be allowed on it under Board of Trade rules? Ans. 112.2 lb. per sq. in.

11. Under American rules, what is the minimum thickness of a smoke flue 30 inches in diameter? Ans. .4 in.

12. Under American rules, what working pressure will be allowed on a properly made smoke flue .42 inch in thickness and 30 inches in external diameter? Ans. 112 lb. per sq. in.

BOILER HEADS AND DRUMHEADS

BOILER HEADS

41. The American rules state in regard to boiler heads: "All heads employed in the construction of cylindrical externally fired boilers for steamers navigating the Red River of the North and rivers whose waters flow into the Gulf of Mexico, shall have a thickness of material as follows:

For boilers having a diameter:

Over 32 inches and not over 36 inches, not less than $\frac{1}{2}$ inch.
Over 36 inches and not over 40 inches, not less than $\frac{9}{16}$ inch.
Over 40 inches and not over 48 inches, not less than $\frac{5}{8}$ inch.
Over 48 inches, not less than $\frac{3}{4}$ inch."

42. In practically all fire-tube boilers, the heads are supported by stays, and the working pressure allowable on the heads is calculated by the rules in Arts. **6** to **18**, which assume proper staying. When the staying is weaker, the working pressure must be reduced until the stress on the stays comes within the legal limit.

DRUMHEADS

43. The heads of steam, water, and mud-drums are either flat, convex (bumped), or concave (dished).

Under American rules, flat drumheads, unstayed, must not exceed 20 inches in diameter; the heads of steam and mud-drums of cylindrical externally fired boilers for steamers navigating the Red River of the North and rivers whose waters flow into the Gulf of Mexico must not be less than $\frac{1}{2}$ inch thick; the flanges of all unstayed flat heads must be made to an inside radius of at least $1\frac{1}{2}$ inches. When the heads exceed 20 inches in diameter, they must be stayed, and their working pressure is determined by the rule for flat surfaces in Art. **6.** For unstayed flat heads, the American rule for finding the working pressure is as follows:

Rule.—*Multiply the constant corresponding to the thickness by the square of the thickness of the material, in sixteenths of an inch, and divide the product by one-half the area of the head, in square inches.*

Or, $$P = \frac{C T^2}{A}$$

in which P = working pressure, in pounds per square inch;
C = 112 for plates $\frac{7}{16}$ inch and under;
C = 120 for plates over $\frac{7}{16}$ inch;
T = numerator of fraction denoting plate thickness, in sixteenths of an inch;
A = one-half the area of the head, in square inches.

EXAMPLE.—What working pressure will be allowed on a unstayed steam drumhead $\frac{3}{4}$ inch thick and 18 inches in diameter?

SOLUTION.— $\frac{3}{4} = \frac{12}{16}$, or $T = 12$.

$$A = \frac{18^2 \times .7854}{2} = 127.2 \text{ sq. in.}$$

$C = 120$. Applying the rule,

$$P = \frac{120 \times 12^2}{127.2} = 135.85 \text{ lb. per sq. in. Ans.}$$

Neither the Board of Trade nor the Canadian rules make provision for unstayed flat heads of steam or mud-drums.

44. The Canadian rules state that in water-tube boilers, drumheads must not be less than $\frac{1}{2}$ inch thick. When stayed by a number of stays, the working pressure on a flat drumhead is found by the rule presented in Art. **9**, both under Board of Trade and Canadian rules.

For the common case where the flat heads of a drum of a water-tube boiler are stayed by a single stayrod in the center of the heads, the Canadian rules specify that the working pressure is to be found as follows, provided the plates of the drum are exposed to the impact of heat or flame (as in the Roberts boiler) and the stayrod is screwed into the heads and fitted with nuts, or well riveted over to form a good head, and the boiler is new:

Rule.—*Multiply 80 by the square of the sum of the thickness of the head, in sixteenths of an inch, and 1. Divide the product by the square of the difference between one-half the inside diameter of the drumhead, in inches, and 1, diminished by 6.*

Or,
$$P = \frac{80\,(T+1)^2}{\left(\frac{d}{2}-1\right)^2 - 6}$$

in which P = working pressure, in pounds per square inch;
T = numerator of fraction denoting the thickness of the head, in sixteenths of an inch;
d = inside diameter of the drumhead, in inches.

EXAMPLE.—The flat head of a steam drum 24 inches inside diameter is stayed by a stayrod in the center and is $\frac{3}{4}$ inch thick; what working pressure will be allowed on it, under Canadian rules?

SOLUTION.— $\frac{3}{4} = \frac{12}{16}$, or $T = 12$. Applying the rule,
$$P = \frac{80 \times (12+1)^2}{\left(\frac{24}{2}-1\right)^2 - 6} = 117.57 \text{ lb. per sq. in. Ans.}$$

45. For unstayed convex heads, that is, heads receiving the steam pressure on their concave side, the American rules prescribe the following rule:

Rule.—*Multiply the thickness of the plate, in inches, by one-sixth of the tensile strength and divide the product by one-half the radius to which the head is bumped. The quotient will be*

the working pressure if the head is single riveted to the drum. When the head is double riveted to the drum, add 20 per cent.; that is, multiply by 1.2.

Or, for single riveting,

$$P = \frac{TS}{R}$$

and for double riveting,

$$P = \frac{TS}{R} \times 1.2$$

in which P = working pressure, in pounds per square inch;
T = thickness of head, in inches;
S = one-sixth of the tensile strength, in pounds per square inch;
R = one-half the radius to which the head is bumped.

To find the radius to which the head is bumped, square one-half the diameter of the head and divide by the height of the bump; to the quotient add the height of the bump and divide the sum by 2.

EXAMPLE.—The head of a steam drum is convex, bumped to a radius of 36 inches, is $\frac{1}{2}$ inch thick, and has a tensile strength of 55,000 pounds per square inch; what working pressure will be allowed on this head: (*a*) if single riveted? (*b*) if double riveted?

SOLUTION.—(*a*) Applying the rule,

$$P = \frac{\frac{1}{2} \times \frac{55000}{6}}{\frac{36}{2}} = 254.63 \text{ lb. per sq. in. Ans.}$$

(*b*) Applying the rule,

$$P = \frac{\frac{1}{2} \times \frac{55000}{6}}{\frac{36}{2}} \times 1.2 = 305.56 \text{ lb. per sq. in. Ans.}$$

46. The Board of Trade rules state: "If dished ends (convex heads) are not equal to the pressure needed when considered as portions of spheres, they should be stayed as flat surfaces."

The general rule for finding the working pressure on a jointless sphere, or a portion of a jointless sphere, is as follows:

Rule.—*Multiply twice the tensile strength of the material, in pounds per square inch, by its thickness, in inches; divide the*

product by the product of the radius of the sphere, in inches, and the factor of safety.

Or,
$$B = \frac{2\,C\,T}{R\,F}$$

in which B = working pressure, in pounds per square inch;
C = tensile strength, in pounds per square inch;
R = radius of sphere, in inches;
F = factor of safety.

The Board of Trade rules do not give the factor of safety to be used in case of convex heads; they imply that it will be 4.5 if the material is tested.

EXAMPLE.—What working pressure will be allowed on a convex steel head $\frac{1}{2}$ inch thick, having a tensile strength of 60,000 pounds per square inch, and bumped to a radius of 30 inches, using a factor of safety of 4.5?

SOLUTION.—Applying the rule,

$$B = \frac{2 \times 60{,}000 \times \frac{1}{2}}{30 \times 4.5} = 444.44 \text{ lb. per sq. in.} \quad \text{Ans.}$$

47. Under Canadian rules, the working pressure on convex heads of steam drums, when not exposed to the impact of heat or flame, is found by the rule in Art. **46**, making $F = 4$ when the heads are pressed into shape by a machine and subsequently annealed, and making $F = 5$ when the heads are worked into shape by hand and subsequently annealed. When a greater pressure is desired than given by the rule, the heads must be stayed as flat surfaces.

48. Neither the Board of Trade nor the Canadian rules make any provision for concave heads, that is, heads that receive the steam pressure on their convex side. The American rules permit a working pressure of .6 that permitted on the same head when used as a convex head; therefore, to find the working pressure on a concave head, apply the rule in Art. **45** and multiply the result by .6.

EXAMPLES FOR PRACTICE

1. Under American rules, what working pressure will be allowed on a flat unstayed head of a mud-drum if the head is $\frac{1}{2}$ inch thick and 10 inches in diameter? Ans. 195.57 lb. per sq. in.

2. What working pressure will be allowed, under Canadian rules, on a flat drumhead 20 inches in diameter and $\frac{13}{16}$ inch thick, properly stayed by one stayrod in its center, and used for a water-tube boiler?
Ans. 209 lb. per sq. in., nearly

3. A convex head is bumped to a radius of 30 inches, is $\frac{3}{8}$ inch thick, and has a tensile strength of 60,000 pounds per square inch; the head being single riveted to the shell, what working pressure will the American rules allow on this head? Ans. 250 lb. per sq. in.

4. If the head in example 4 were concave, what pressure would the American rules allow on it? Ans. 150 lb. per sq. in.

5. A convex head for the steam drum of a fire-tube boiler is bumped to a radius of 24 inches and is worked into shape by hand; it is made of wrought iron having a tensile strength of 50,000 pounds per square inch, and is $\frac{3}{8}$ inch thick. What working pressure will be allowed on this head under Canadian rules? Ans. 312.5 lb. per sq. in.

PIPES AND SAFETY VALVES

PIPES

49. Under American rules, the working pressure allowable on wrought-iron or steel pipes is found as follows:

Rule.—*Multiply 10,000 by the thickness of the pipe, in inches, diminished by .125, and divide the product by the inside diameter, in inches.*

Or,
$$P = \frac{10{,}000\ (T - .125)}{D}$$

in which P = working pressure, in pounds per square inch;
T = thickness of pipe, in inches;
D = inside diameter, in inches.

Example.—What working pressure will be allowed on a main steam pipe 12 inches in inside diameter and $\frac{3}{8}$ inch thick, under American rules?

Solution.—Applying the rule,
$$P = \frac{10{,}000 \times (\frac{3}{8} - .125)}{12} = 208.33 \text{ lb. per sq. in.} \quad \text{Ans.}$$

50. The Canadian rules do not prescribe any formula for finding the working pressure allowable on wrought-iron

or steel pipes; under Board of Trade rules, it is found as follows:

Rule.—*Multiply 6,000 by the thickness of the pipe, in inches, and divide the product by the inside diameter, in inches.*

Or,
$$P = \frac{6{,}000\ T}{D}$$

in which the letters have the same meaning as in Art. 49.

EXAMPLE.—Under Board of Trade rules, what working pressure will be allowed on a wrought-iron steam pipe 12 inches in inside diameter and $\frac{3}{8}$ inch thick?

SOLUTION.—Applying the rule,
$$P = \frac{6{,}000 \times \frac{3}{8}}{12} = 187.5 \text{ lb. per sq. in. Ans.}$$

51. The Canadian rules do not contain a rule for finding the working pressure allowable on copper pipes; the American and Board of Trade rules give the same rule, differing only in the value of the constant.

Rule.—*Multiply 8,000 (under American rules) or 6,000 (under Board of Trade rules) by the thickness of the pipe diminished by $\frac{1}{16}$ inch. Divide the product by the inside diameter of the pipe.*

Or, American rules,
$$P = \frac{8{,}000\ (T - \frac{1}{16})}{D}$$

and, Board of Trade rules,
$$P = \frac{6{,}000\ (T - \frac{1}{16})}{D}$$

in which the letters have the same meaning as in Art. 49.

EXAMPLE.—A copper feedpipe $\frac{3}{16}$ inch thick has a diameter of 4 inches; what working pressure will it be allowed, under American rules?

SOLUTION.—Applying the rule,
$$P = \frac{8{,}000 \times (\frac{3}{16} - \frac{1}{16})}{4} = 250 \text{ lb. per sq. in. Ans.}$$

SAFETY-VALVE AREA

52. Under American rules, the area of a safety valve, in square inches per square foot of grate surface, is to be found by the rule in this article. Obviously, the result of the rule must be multiplied by the number of square feet of grate surface of the boiler in order to obtain the area of the safety valve required. When the diameter exceeds 6 inches, it is customary to use two valves whose combined area will equal that calculated. If the calculation gives an odd size of safety valve, the next larger standard size is chosen.

Rule.—*Multiply the constant .2074 by the number of pounds of water evaporated per hour per square foot of grate surface. Divide the product by the absolute boiler pressure, taken as the gauge pressure plus 15.*

Or,
$$a = \frac{.2074\,W}{P}$$

in which a = area of safety valve, in square inches, per square foot of grate surface;

W = pounds of water evaporated per square foot of grate surface per hour;

P = working pressure, in pounds per square inch, + 15.

EXAMPLE.—A boiler is estimated to burn 12 pounds of coal per square foot of grate surface per hour, and to evaporate 9 pounds of water per pound of coal; the boiler having 30 square feet of grate surface, what area should the safety valve have, under American rules? The boiler pressure is 150 pounds per square inch.

SOLUTION.— $W = 12 \times 9 = 108$, $P = 150 + 15 = 165$. Applying the rule,

$$a = \frac{.2074 \times 108}{165} = .1358 \text{ sq. in.}$$

Then, area of safety valve for the boiler is

$$.1358 \times 30 = 4.074 \text{ sq. in. Ans.}$$

Under American rules prior to June 1, 1904, lever safety valves were required to have an area of 1 square inch for every 2 square feet of grate surface, and spring-loaded safety valves, 1 square inch for every 3 square feet of grate surface.

For water-tube, coil, and sectional boilers carrying a pressure in excess of 175 pounds per square inch, the requirements were 1 square inch of safety-valve area for every 6 square feet of grate surface.

53. The Canadian rules prohibit the use of a safety valve smaller than 1 inch diameter; under Board of Trade rules, the diameter must not be less than 2 inches.

Under both Board of Trade and Canadian rules, the area of a safety valve, in square inches per square foot of grate surface, for boilers worked under natural draft, is found by the rule given in this article. Obviously, the result of the rule must be multiplied by the number of square feet of grate surface of the boiler to find the safety-valve area for that boiler.

Rule.—*Divide 37.5 by the absolute boiler pressure, taken as the sum of the gauge pressure and 15.*

Or,
$$a = \frac{37.5}{P}$$

in which the letters have the same meaning as in Art. **52.**

EXAMPLE.—What area of safety valve is required, under Canadian and Board of Trade rules, for a boiler worked under natural draft at a gauge pressure of 150 pounds per square inch, the boiler having 30 square feet of grate surface.

SOLUTION.—Applying the rule,
$$a = \frac{37.5}{150 + 15} = .227 \text{ sq. in.}$$
Then, area of safety valve is
$$.227 \times 30 = 6.81 \text{ sq. in. Ans.}$$

54. For boilers worked under forced draft, the safety-valve area, in square inches per square foot of grate surface, is found as follows, under Board of Trade rules:

Rule.—*Multiply the area of safety valve per square foot of grate surface calculated for natural draft (by the rule in Art. 53) by the estimated coal consumption per square foot of grate surface per hour, in pounds, and divide the product by 20.*

Or,
$$A = \frac{a\,C}{20}$$

in which A = area of safety valve per square foot of grate surface for forced draft, in square inches;

a = area of safety valve per square foot of grate surface for natural draft, in square inches;

C = coal consumption, in pounds per hour per square foot of grate surface.

EXAMPLE.—Under Board of Trade rules, what area of safety valve will be required for a boiler having a grate surface of 42 square feet, burning 40 pounds of coal per square foot of grate surface per hour under forced draft, and carrying a steam pressure of 180 pounds per square inch?

SOLUTION.—Applying the rule in Art. **53**,

$$a = \frac{37.5}{180 + 15} = .192 \text{ sq. in.}$$

Applying the rule in Art. **54**,

$$A = \frac{.192 \times 40}{20} = .384 \text{ sq. in.}$$

Then, area of safety valve is

$$.384 \times 42 = 16.128 \text{ sq. in.} \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE

1. Under American rules, what working pressure will be allowed on a wrought-iron steam pipe $\frac{1}{4}$ inch thick and 7 inches in inside diameter? Ans. 178.57 lb. per sq. in.

2. What working pressure will be allowed, under Board of Trade rules, on the pipe in example 1? Ans. 214.29 lb. per sq. in.

3. A copper pipe 6 inches in inside diameter has a thickness of $\frac{1}{4}$ inch; what working pressure will be allowed on it: (*a*) under American rules? (*b*) under Board of Trade rules?

Ans. { (*a*) 250 lb. per sq. in.
(*b*) 187.5 lb. per sq. in.

4. A boiler is estimated to evaporate 120 pounds of water per hour per square foot of grate surface, which measures 60 square feet; what area of safety valve is required for the boiler, under American rules, if the working pressure is to be 105 pounds per square inch by the gauge? Ans. 12.44 sq. in.

5. A boiler having 80 square feet of grate surface is to be worked at a gauge pressure of 160 pounds per square inch under natural draft; what area of safety valve is required, under Canadian rules? Ans. 17.14 sq. in.

6. Under Board of Trade rules, what safety-valve area is required for a boiler burning 36 pounds of coal per square foot of grate surface per hour under forced draft, the boiler pressure being 185 pounds per square inch by the gauge, and the grate surface 60 square feet?

Ans. 20.25 sq. in.

PROPULSION OF VESSELS

PROPELLING INSTRUMENTS

INTRODUCTION

METHODS OF PROPULSION

1. A steam vessel may be propelled either by a stream of water, caused by suitable means to flow in a direction opposite to that in which it is desired to propel the vessel; or it may be pulled along a stationary chain or cable lying on the bottom of the river, the chain passing around a drum situated within the vessel and actuated by suitable machinery. Chain propulsion is used to some extent in Europe, but since it does not possess any special features calling for a description of the system, it will not be treated here. In the first method of propulsion mentioned, the stream of water projected from the vessel propels the vessel by its reaction.

In practice, a stream of water may be projected from a vessel in three ways: (1) By means of one or more paddle wheels; when there is only one wheel, it is usually situated at the stern of the vessel; when there are two, they are placed amidships, or nearly so. (2) By means of one or more screws situated either at the stern or at both the bow and the stern. (3) By means of a pump placed within the vessel.

In the last case, a stream of water is led to the pump by suitable piping, and is ejected under pressure from orifices located in the sides of the vessel. A vessel propelled in this manner is known as a *jet propeller*. Owing to practical difficulties, jet propulsion has never come into general use, and, hence, will not be treated here.

SLIP

2. Action of Propelling Instrument.—When a paddle wheel or screw serving as a propelling instrument is revolved, it tends to force backwards a certain quantity of water; the inertia of the water opposes this effort, and, by virtue of the reaction thus created, the vessel is propelled. When the engines are first started, the propelling instrument revolves for a certain time before the inertia of the vessel is overcome, and during this short space of time the water that is driven backwards has nearly the same velocity with respect to the main body of water that it has to the ship. Suppose after the ship is under way, that the instrument maintains the same speed of revolution and the speed of the ship gradually increases; then, as the stream of water forced back maintains a constant velocity relative to the ship, it is seen that its velocity relative to the main body of water is gradually decreasing.

The speed of the ship will gradually increase up to a certain point, due to the force of propulsion, and then remain at that speed—that is, provided the velocity of the issuing stream remains the same. The whole of the propelling effort is now absorbed in overcoming the resistance of the air, skin friction, inertia of the water, etc.

3. Measurement of Ship's Speed.—The speed of a vessel may be measured in two ways—by its motion through water and by its motion over ground. The latter, which is the actual speed in reference to a fixed point on land, is obtained from the former by allowing for the motion of the water in which the vessel floats. There may be a decided difference in speed measured in the two ways mentioned, as the following consideration will show. Let the distance between two ports be 1,000 miles, and let the ship cover this distance in 100 hours. Then, its speed in relation to a fixed point of the earth (the port of departure) is $\frac{1000}{100} = 10$ miles per hour. Now, assume that the vessel is again leaving the same port of departure bound for the

same port of destination and over the same route, but that on leaving port it enters a current setting in the same direction as the ship is advancing, and running 2 miles per hour. Then, in order to reach the port of destination in the same time as before, that is, in 100 hours, its speed per hour in regard to the point of departure must be 10 miles; but, owing to being in a current advancing the vessel 2 miles per hour, its speed through the surrounding water must be $10 - 2 = 8$ miles per hour. This shows that there may be a difference between the speed of a ship through the surrounding water, that is, the speed with which the surrounding water may be conceived to move past the ship, and the speed of a ship relative to a fixed point of the earth. It is of the utmost importance that this be kept in mind. In considering the speed of a vessel from the propulsion point of view, its speed in respect to the surrounding water must be taken.

4. Wake.—By reason of friction, a moving vessel will drag with it some water, which forms a casing, as it were, the different parts of which have different velocities. Thus, the water directly in contact with the part of the vessel below the water-line will have a velocity probably nearly equal to that of the ship, while particles of water several inches from the ship will have a much lower velocity, and so on, until, finally, those some distance from the vessel will have no motion in relation to the surrounding water. The water thus dragged along, with its velocity modified by other conditions, will collect near the stern of the vessel in the form of a stream moving in the same direction as the vessel; this stream is called the **wake**. When standing at the stern of a ship and looking at the wake, it appears to recede from the ship; this, of course, is due to the fact that the vessel moves forwards faster than the wake. If the vessel could be instantly brought to rest, the wake would flow toward the ship; in other words, its motion in relation to the ship is always forwards. The velocity of the wake, in relation to the surrounding water and the ship, is greatly influenced by the form of the vessel below the water-line,

one with a blunt stern imparting a greater velocity to its wake than one with a fine after body. It is readily seen that the creation of a wake reduces the force available for giving speed to the vessel, as a considerable part of the total force exerted by the engine is expended in dragging the water forming the wake.

5. True Slip.—In considering the speed of a stream projected by a propelling instrument from a vessel in motion, it must be borne in mind that while the stream is propelled astern the vessel is advancing. Since the stream must move astern faster than the vessel advances, the rearward speed of the stream in relation to a fixed point of the water some distance astern of the ship, as a floating piece of wood, will be the difference between the speed of the vessel in relation to the piece of wood and the rearward speed of the stream in relation to the vessel. If the propelling instrument, as for instance the paddle wheels of a side-wheel steamer, does not work in a wake, the speed of the vessel, in relation to a fixed point of the water astern, may be conceived to be the average speed with which the water is fed to the propelling instrument. When the propeller works in a wake, however, which, as previously stated, has a forward motion, the speed with which water is fed to the propelling instrument is reduced thereby and it becomes equal to the difference between the forward speed of the vessel in relation to a fixed point of the water clear of the wake and the wake velocity. Thus, if the speed of the ship is 15 miles per hour and a wake that has a forward velocity of 3 miles per hour collects at the stern, the speed with which the water is fed to a screw propeller is $15 - 3 = 12$ miles per hour. The difference between the speed with which water is fed to the propelling instrument and the speed with which it is projected astern, both speeds being measured in relation to the vessel, is called the **true slip**, and also the **real slip**, of the stream.

6. Assume that a vessel whose propelling instrument is working in water clear of the wake is descending a river,

which means that it floats in a favoring current. The speed of the ship in relation to its port of departure will be increased by the current; that is, the speed of the ship through the water remaining the same, the advance of the ship in relation to the port of departure will be the sum of the speed through the water and the speed of the current in relation to the port of departure. Letting a denote the speed of the ship through the water and c the speed of the current, the speed of the ship in relation to the port of departure is $a + c$. Likewise, if d denotes the speed of the ship in relation to the port of departure, its speed through the water will be $d - c = a$. If b denotes the speed of the stream projected from the vessel, measured in relation to the vessel, the true slip, measured in relation to the port of departure, is $b - (d - c)$, which is the same as $b - a$.

Suppose that the vessel is moving through still water, but that a wake is following the ship and that the propelling instrument is working in this wake. This wake can be likened to the current previously mentioned, and hence its forward speed can be denoted by c. Then, if d denotes the advance (the speed) of the ship in relation to a fixed point in the water in which it floats (this point is the port of departure of the previous case), the true slip, as before, is $b - (d - c)$; or if a denotes the advance of the vessel in relation to a fixed point of the wake, the true slip is $b - a$.

7. A force must act continuously to propel a vessel through the water; this force is the reaction of the stream projected astern by the propelling instrument and is proportional to the true slip, the cross-sectional area of the stream remaining the same. From this statement, it follows that without true slip there can be no reaction, that is, no propulsive force; in other words, propulsion of a vessel without real slip is an impossibility.

8. Apparent Slip.—In practice, it is very inconvenient and exceedingly difficult to measure either the wake velocity or the speed of the ship in relation to the wake, but it is a very simple matter to measure the speed of the vessel in relation

to a fixed point in the water clear of the wake by means of an instrument called a **log.** This instrument consists of three parts: the log chip, the log line, and the log glass. The chip, which is a triangular piece of light wood attached to the line, is thrown overboard; as it strikes the water it soon virtually ceases to partake of the ship's onward motion and becomes stationary; the distance of this stationary object from the ship is then measured after a certain interval of time has passed and from this the approximate rate of speed is ascertained, the log glass defining the interval of time. The difference between the speed of the stream projected by the propelling instrument and the speed of the ship thus found, that is, $b - d$, Art. **6,** is taken as the slip. When calculated in this manner it obviously is not the same as the true slip; it is called the **apparent slip.** Conditions are possible under which the true slip and apparent slip may have the same numerical value; this, however, does not change the fact that they are separate quantities.

9. Negative Slip.—It is perfectly possible for a propelling instrument to have an apparent slip of zero, and under certain conditions calculation may even show the apparent slip to be less than zero; it is then called **negative slip.** This arises from failing to take into account either the wake velocity or the velocity of the vessel aided by a favorable wind, in case the speed of the vessel is measured, as it should be in considering propulsion problems, in reference to a fixed point of the water clear of the wake. When the speed of the vessel is measured in respect to a fixed point of the earth, as its port of departure, the existence of a favoring current is an additional cause that may produce a negative apparent slip, either by itself or in conjunction with the wake velocity and wind effect.

10. In algebra, a number greater than zero is called a *positive number*, and a number less than zero is called a *negative number*. It is not possible to arithmetically subtract a positive subtrahend from a smaller positive minuend; this is easily done algebraically, however, with the aid of the following rule.

Rule.—*To algebraically subtract one positive number from a smaller positive number, subtract the smaller number from the larger one and prefix the minus sign to the difference to indicate that it is negative.*

EXAMPLE 1.—Subtract 12 from 8.

SOLUTION.—
$$\begin{array}{r} 12 \\ \underline{8} \\ -4 \end{array} \text{ Ans.}$$

A negative number can be divided by a positive number as in arithmetic; a minus sign is prefixed to the quotient to indicate that the dividend was a negative number.

EXAMPLE 2.—Divide −144 by 6.

SOLUTION.—
$$\begin{array}{l} 6\,)\,-144\,(\,-24 \text{ Ans.} \\ \quad\;\; \underline{12} \\ \quad\;\;\; 24 \\ \quad\;\;\; \underline{24} \end{array}$$

11. Assume a side-wheel steamer to be descending a river whose current is running 5 miles per hour, that its speed through the water is 10 miles per hour, as shown by the log, and that the paddle wheels project streams to the rear at the rate of 12 miles per hour. In a side-wheel steamer, the propelling instruments are practically clear of the wake, so that the problem under consideration is not complicated by having to take a wake velocity into account. The speed with which the ship advances in respect to the port of departure is $10 + 5 = 15$ miles per hour; the real, or true, slip is $12 - 10 = 2$ miles per hour, but the apparent slip is $12 - 15 = -3$ miles per hour; that is, apparently, the ship is moving faster than the stream projected by the propelling instrument. Obviously, the presence of negative slip, in this particular case, is due to taking the ship's speed in relation to the port of departure instead of in reference to the water in which the paddle wheels work. Taking the ship's speed in reference to the water, the apparent slip for this case $12 - 10 = 2$ miles per hour, or the same as the true slip, as here the wake velocity does not enter the problem.

12. The influence of the wake velocity on the apparent slip when the ship's speed is taken from the log is similar to that of a favorable current in that case where the speed of a ship is reckoned from its advance from the port of departure. The log does not show the ship's speed in relation to the wake, but in relation to a fixed point in the water clear of the wake (the port of departure in case of the paddle-wheel steamer mentioned in Art. **11**), and hence shows a higher speed in relation to the water in which the propelling instrument works than really exists. Consequently, when the propelling instrument works in water having a high forward velocity, a neglect to take this forward velocity, or wake, into account may result in an extremely small or even negative apparent slip.

13. The apparent slip can be reduced, or even made negative, by a wind helping the vessel along. Thus, if a vessel advances 15 miles per hour, as shown by the log, which means in relation to a fixed point of the water clear of the wake, of which speed $13\frac{1}{2}$ miles per hour is due to the propelling instrument and $1\frac{1}{2}$ miles per hour due to the wind, and if the propelling instrument projects a stream at the rate of $14\frac{1}{2}$ miles per hour, the apparent slip is $14\frac{1}{2} - 15 = -\frac{1}{2}$, that is, negative. Taking into consideration the speed produced by the wind, however, the apparent slip would be $14\frac{1}{2} - (15 - 1\frac{1}{2}) = 1$ mile per hour.

14. When a calculation for apparent slip—where the ship's velocity is taken in relation to the water through which it advances, that is, from a fixed point clear of the wake—shows an extremely small apparent slip, or even a negative slip, it proves usually that one or both of two conditions exist. These are an abnormally high wake velocity, and an increase in the speed of the ship by the action of a favorable wind. An abnormally high apparent slip denotes that the propelling instrument is unsuitable for the conditions under which the vessel is propelled at the time for which the calculation was made; if the apparent slip has become abnormally high in comparison to that usually

existing, it tends to show that the resistance of the ship has been increased in some manner, as by a head-wind or head-sea or by a foul bottom, or that the efficiency of the propelling instrument has suddenly been decreased, as by the breaking of one or more blades of a screw propeller.

15. Negative apparent slip is often found in stern-wheel steamers and in screw steamers having a blunt stern. Negative slip due to a high wake velocity is not observed in side-wheel steamers, as the paddle wheels are located clear of the wake; these steamers can show a negative slip due to a favorable current or wind action, however. The former, as is also the case with stern-wheel and screw steamers, can occur when the speed of the ship is taken in reference to a fixed point of the ground instead of in reference to the water in which the ship floats, as it should be; a negative apparent slip can also occur when the speed of the ship is properly taken in reference to the water in which it floats, but no allowance is made for the increase of speed due to the wind. The same remarks also apply to an abnormally low apparent slip.

16. The existence of negative apparent slip, when traced to a high wake velocity, shows a poor efficiency of propulsion for the existing conditions, as measured by the horsepower developed for the speed. The fitting of a propelling instrument giving a positive apparent slip will generally greatly improve the efficiency of propulsion.

17. Rules for Slip.—It is customary to express slip in per cent. of the velocity of the stream projected by the propelling instrument. This may be done by the following rules:

Rule I.—*To find the true slip, from the velocity of the stream projected by the propelling instrument subtract the velocity with which the water is fed to the propelling instrument, both velocities being expressed in any convenient, but the same, measure of time and distance. Divide the difference by the velocity of the stream projected by the propelling instrument.*

Or,
$$S_t = \frac{V - V_1}{V}$$

where S_t = true slip in per cent., expressed decimally;
V = velocity of stream projected by propelling instrument, in relation to vessel;
V_1 = velocity of water fed to propelling instrument; that is, speed of vessel in relation to the surrounding water diminished by the wake velocity at the point where the propelling instrument is located, for a vessel under way.

EXAMPLE 1.—Find the true slip when the speed of the vessel, by the log, is 12 miles per hour; the wake velocity 3 miles per hour; and the stream is projected by the propelling instrument at 13 miles per hour.

SOLUTION.—Applying rule I,
$$S_t = \frac{13 - (12 - 3)}{13} = .3077 = 30.77 \text{ per cent. Ans.}$$

Rule II.—*To find the apparent slip in reference to the ship's motion through the water, from the velocity of the stream projected by the propelling instrument subtract the speed of the vessel in relation to the water. Divide the difference by the velocity of the stream projected by the propelling instrument. The velocities may be expressed in any convenient measure of time and distance, but the same measure must be used for both.*

Or,
$$S_a = \frac{V - V_2}{V}$$

where S_a = apparent slip in reference to the ship's motion through the water, expressed decimally and in per cent.;
V = velocity of stream projected by the propelling instrument;
V_2 = speed of vessel in relation to the water, as shown by the log.

EXAMPLE 2.—Find the apparent slip for the vessel in example 1.

SOLUTION.—Applying rule II,
$$S_a = \frac{13 - 12}{13} = .0769 = 7.69 \text{ per cent. Ans.}$$

18. Apparent slip when calculated properly, that is, in reference to a fixed point of the water clear of the wake,

is a measure of the efficiency of propulsion of that vessel when considered in conjunction with the true slip. It should never be construed to be a measure of the efficiency of propulsion if considered only by itself. Experience has shown that a very low apparent slip by itself may be an indication of poor efficiency; in that case, it will be combined with a high true slip. A low apparent slip coupled with a low true slip, however, indicates a high efficiency, while a high apparent slip and high true slip indicate a poor efficiency for the existing conditions. Likewise, a high apparent slip and a low true slip indicate a poor efficiency of propulsion for the existing conditions. This fact is illustrated by considering a vessel moored to the dock, with the engine working ahead. As the vessel does not advance at all, the apparent slip is 100 per cent.; the true slip, as measured by the difference in velocity of the stream projected by the propeller and the speed of the water fed to it, may be extremely low, say 1 per cent. Then, for this case, the efficiency of propulsion, if perfection is taken as 1, is $1 - 1 = 0$; while the efficiency of the propeller is $1 - .01 = .99$.

19. It is customary among writers on marine propulsion to refer to apparent slip simply as slip; when the true slip is meant, it is usual to qualify the term slip by prefixing the word true or real, that is, to call it distinctly the true slip or the real slip.

PADDLE WHEELS

DEFINITIONS

20. The **paddle wheel**, in its simplest form, consists of two rings, concentric with the axis of the shaft and lying in planes perpendicular to it, that are secured, by arms, to a hub keyed to the shaft. At the outer edges of the rings, and placed between them, are the **buckets** (sometimes called **floats**), which are either flat wooden boards, generally elm, or iron plates; they are situated at equal distances apart, in planes passing through the axis of the shaft. A paddle

wheel with the buckets placed thus is known as a *radial wheel*. The wheel, or wheels, is attached to the vessel in such a position that the lower part is immersed in water to a certain depth, which is usually spoken of as the **dip** of the wheel. The vertical distance from the inner edge of the buckets to the surface of the water, is called the **immersion of the buckets.**

SLIP OF PADDLE WHEEL

21. In order to find the velocity at which a stream of water is projected by a paddle wheel, it is necessary to first find that point of the wheel at which its whole action on the water may be assumed to be concentrated. This point is known as the **center of pressure,** often called the **center of action** of the wheel; and twice the distance of this point from the outer edge of the buckets subtracted from the diameter of the wheel (the diameter to be measured from outer edge to outer edge of buckets) constitutes the **effective diameter** of the wheel.

The effective diameter may be determined approximately by the following rule:

Rule.—*Multiply one-third the mean depth of the buckets wholly immersed by the number of buckets so immersed. To this product, add the product of one-third the mean depth of the buckets partly immersed and their number. Divide the sum of the two products by the number of the buckets partly and wholly immersed. The quotient will be the distance of the center of pressure from the outer edge of the buckets. The effective diameter can then be found by subtracting twice the distance of the center of pressure from the outer edge of the buckets from the outside diameter of the wheel.*

Or, $$D_e = D - 2 \times \frac{ab + cd}{3(b + d)}$$

where P = distance of center of pressure from outer edge of buckets, in inches;

a = mean depth of buckets wholly immersed, in inches;

b = number of buckets wholly immersed;
c = mean depth of buckets, partly immersed, in inches;
d = number of buckets partly immersed;
D = diameter of wheel, measured over outer edge of buckets;
D_e = effective diameter.

To find the mean depth of the buckets, and also the number of buckets wholly and partly immersed, draw the wheel to any convenient scale, taking care to draw the buckets in their true positions. Also, draw a line representing the surface of the water, at a distance from the outer edge of the lowest bucket equal to the dip. Then the number of buckets wholly and partly immersed will be seen at a glance. To find the mean depth of the buckets, measure the depth of each bucket wholly immersed (not the depth to which each bucket is immersed) to the same scale the wheel was drawn, and perpendicular to the surface of the water. Add the depths of the different buckets together, and divide the sum by the number of buckets wholly immersed. For those partly immersed, which hardly ever will be more than two, measure the perpendicular distance between the lower edge of the bucket and the surface of the water. Add the distances together, and divide by the number of buckets.

EXAMPLE.—A paddle wheel 36 feet in diameter has, at a certain dip, seven buckets of a mean depth of 24 inches wholly immersed, and one bucket immersed to a depth of 15 inches; find the effective diameter of the wheel.

SOLUTION.—Applying the rule,

$$D_e = 36 \times 12 - 2 \times \frac{24 \times 7 + 15 \times 1}{3(7 + 1)} = 416.75 \text{ in. Ans.}$$

Having determined the effective diameter, to determine the theoretical velocity of the stream projected by the wheel, find the velocity of a point on the circle having a diameter equal to the effective diameter of the wheel, expressing its velocity in the same terms in which the speed of the vessel is expressed. For example, taking the wheel in the above example, its circumference will be 3.1416 × 416.75 = 1,309.26

inches. Assuming the revolutions to be 20 per minute, the speed of a point on the effective diameter circle will be, in feet per second, $\frac{1,309.26 \times 20}{12 \times 60} = 36.37$ feet. This is, theoretically, the velocity of the stream projected by the wheel; and, knowing the velocity of the vessel, the slip may be found by rule I or rule II, Art. 17.

RADIAL PADDLE WHEEL

22. In Fig. 1, a **radial paddle wheel** is shown in diagrammatic form, where A represents the shaft; B, B_1, B_2,

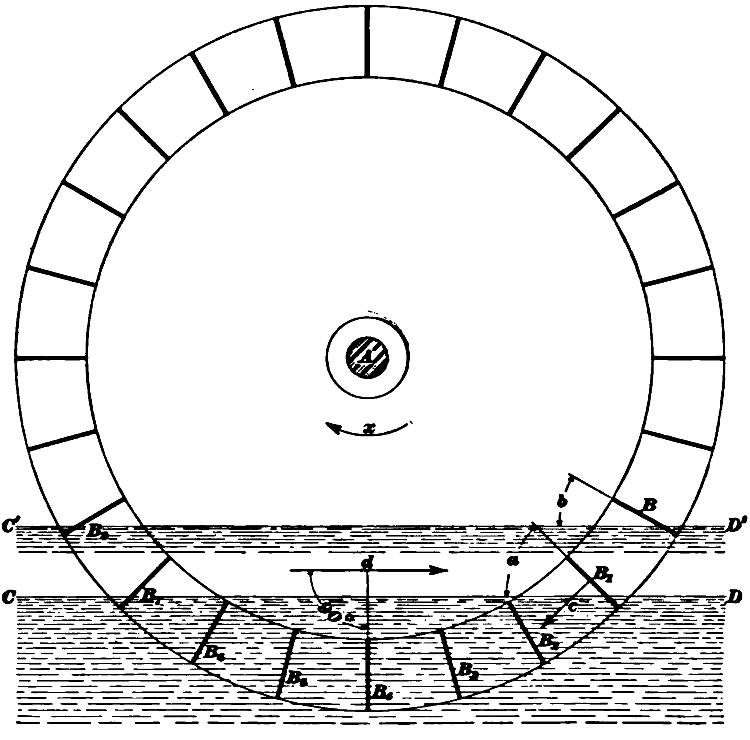


FIG. 1

etc. represent the buckets. Assume the wheel to revolve in the direction of the arrow x; also, assume the line CD to

represent the surface of the water. Then, the direction in which the vessel moves, which is evidently parallel to the surface of the water, is shown by the arrow *d*. It will be seen that the bucket B_1, just entering the water, enters at an angle, shown by the arc *a*, with the surface. This angle is known as the **angle of incidence.** The effect of this is that the bucket, instead of driving a body of water straight astern, drives it in a direction perpendicular to the surface of the bucket, as indicated by the arrow *c*; in other words, the bucket depresses the water. The power consumed in depressing the water is wasted. Evidently, the stream of water projected by the propelling instrument will have the maximum propelling effect if projected straight astern, in a direction parallel to the surface of the water. This is proved by the following simple experiment: Place some heavy weight, weighing, say, about 100 pounds, on the floor. Try to slide it along the floor by pushing against the end of a board placed against the weight, and held at an angle of about 45° with the floor. The chances are that it will not be possible to move it. Depress that end of the board against which the push is exerted, and push just as hard as in the first place. It will now be found possible to move the weight, and it will also be found that the more the free end of the board is depressed, the less will be the power required to move the weight. This proves that the nearer the direction of a force is to the direction in which a weight (as the vessel) is to be moved, the less will be the amount of power required. From this, it follows that the nearer the angle of incidence is to 90°, the more efficiently will the power be applied. By reference to Fig. 1, it will be seen that it is not alone through the action of the bucket B_1 that power is lost, but also that a further loss of power is due to the oblique action of the buckets B_2 and B_3. The only bucket that is acting at its maximum efficiency is B_4, the surface of which is perpendicular to the direction in which the vessel moves. It will be observed that the buckets B_5, B_6, and B_7 tend to elevate a body of water; this causes a loss of power equal to that caused by the action of the buckets

B_1, B_2, and B_3. The sum of the two losses is called the *loss of effect due to oblique action of the buckets*, and is a defect inseparable from the employment of a radial paddle wheel.

Assume the vessel to be loaded until the surface of the water is at $C' D'$. It will be seen that the angle of incidence b is less than it was previously; hence, more power will be uselessly expended. From the foregoing explanations, the following conclusion is drawn: The greater the dip of a paddle wheel, the greater will be the loss of power due to oblique action of the buckets.

FEATHERING PADDLE WHEEL

23. In order to prevent the loss of power incidental to the use of radial buckets, a paddle wheel in which, by a suitable mechanism, the buckets are forced to enter the water perpendicularly, or nearly so, is often used. Such a wheel, which is known as a **feathering paddle wheel,** is shown in Fig. 2. The buckets B, B_1, . . . B_9, turn on pins fixed in brackets a attached to the arms A of the wheel; they are free to move on axes parallel to the axis of the shaft. To the outboard end of each bucket, a lever L is rigidly attached; in order to control the buckets, the extremity of each lever is connected to the eccentric strap F by means of a radius rod r, which is pivoted to the strap as well as to the lever. An eccentric pin c is placed at a distance d ahead of the shaft. The eccentric pin is stationary, but the eccentric strap is free to revolve on the pin. The pin is supported by means of the bracket E, which, in turn, is bolted to the sponson beam G. To give motion to the eccentric strap, it is attached to one of the bucket levers by means of the kingrod H. The kingrod, which is rigidly fastened to the eccentric strap, is pivoted to the bucket lever. As the wheel revolves, each bucket in its turn, on entering the water, assumes the position in which the bucket B is shown, and, in passing around with the wheel, assumes the positions of the buckets B_1, B_2, . . . B_9.

It will be seen at a glance that, for wheels of equal diameter and equal dip, the angle of incidence is much larger

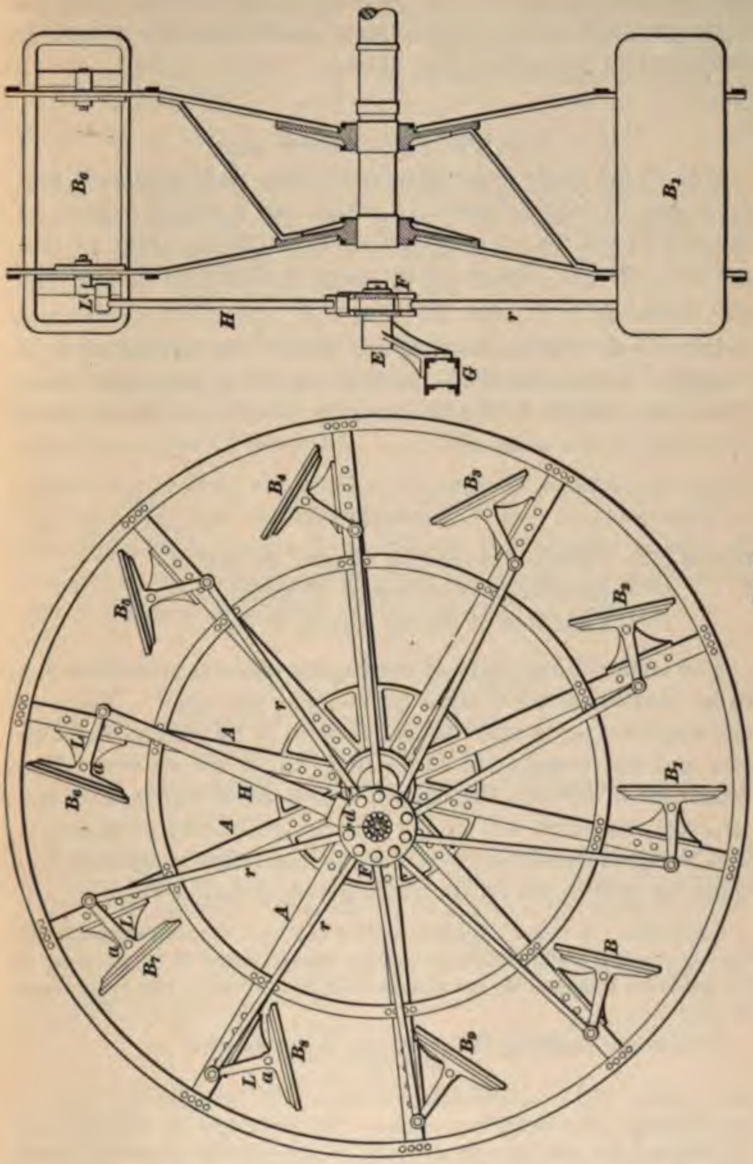


FIG. 2

with the feathering than with the radial paddle wheel. Hence, a larger proportion of the power applied to the wheel is expended in propelling the vessel.

ROLLING CIRCLE

24. The circle concentric with the paddle wheel, any point on the circumference of which has a velocity equal to the velocity of the vessel is called the **rolling circle of the paddle wheel.** Its diameter may be found by the following rule:

Rule.—*To find the diameter, in feet, of the rolling circle of a paddle wheel, divide the distance moved by the vessel in a given time, in feet, by 3.1416 times the number of revolutions of the wheel in the same time.*

Or,
$$D = \frac{S}{3.1416\,R}$$

where S = distance moved by vessel, in feet;
R = number of revolutions of wheel;
D = diameter of rolling circle, in feet.

The term rolling circle of the paddle wheel is an expression of no particular value, although considerably used. The true and apparent slips may be determined if the effective diameter and the diameter of the rolling circle are known. The difference in the two diameters, expressed in per cent. of the effective diameter, will be the percentage of true or apparent slip, and will be found to be the same, all data remaining the same, as that found by the rules given in Art. **17.**

EXAMPLE.—A vessel advances 1,570.8 feet in 1 minute, as shown by the log, during which time the paddle wheels make 25 revolutions; if the effective diameter of the wheels is 25 feet, what is the percentage of slip?

SOLUTION.—Applying the rule given in this article,
$$D = \frac{1,570.8}{3.1416 \times 25} = 20 \text{ ft.},$$
the diameter of the rolling circle. The difference in diameters is $25 - 20 = 5$ ft.; and this, in per cent. of the effective diameter equals $\frac{5}{25} = .2 = 20$ per cent., the apparent slip. Ans.

The velocity of the stream projected by the wheels is $25 \times 25 \times 3.1416 = 1,963.5$ ft. Substituting values in rule II, Art. **17**,

$$S_a = \frac{1,963.5 - 1,570.8}{1,963.5} = .2 = 20 \text{ per cent., the same as above.}$$

SIZE OF PADDLE WHEELS

25. Diameter.—The effective diameter of a paddle wheel is found by the rule below. In order to apply this rule, an apparent slip has to be assumed, and the velocity of the ship in relation to the water it floats in, as well as the number of revolutions, has to be known. The apparent slip for a radial wheel varies from 15 to 30 per cent., the lower value occurring with buckets of ample area, and averages about 25 per cent.; the apparent slip of a feathering paddle wheel averages 15 per cent.

Rule.—*To find the effective diameter of a paddle wheel, in feet, multiply the difference between 1 and the percentage of apparent slip, expressed decimally, by the proposed number of revolutions per minute and by 3.1416. Divide the speed of the ship per minute in relation to the water by this product.*

Or, $$D_e = \frac{V_s}{(1 - S_a)\, 3.1416\, N}$$

where D_e = effective diameter, in feet;
S_a = apparent slip, in per cent.;
V_s = velocity of ship, in feet per minute;
N = number of revolutions per minute.

EXAMPLE.—A vessel is to make 10 statute miles per hour through the water when the wheels make 30 revolutions per minute; assuming a slip of 25 per cent., what should be the effective diameter of the paddle wheels?

SOLUTION.— 10 statute mi. per hr. $= \frac{10 \times 5,280}{60} = 880$ ft. per min. Applying the rule given,

$$D_e = \frac{880}{(1 - .25) \times 3.1416 \times 30} = 12.45 \text{ ft. Ans.}$$

26. Area and Number of Buckets.—For radial and feathering paddle wheels, when the ship is a side-wheel steamer, Seaton recommends that the area, in square feet, of one bucket and the number be as follows:

Rule.—*To find the area of one bucket for the paddle wheels of a side-wheel steamer, divide the indicated horsepower of the propelling machinery by the effective diameter of the wheel, in feet. For a radial paddle wheel, multiply the quotient by .25 for slow boats, and by .175 for fast-running light steamers, choosing a value between the two given as judgment indicates it should be varied. For a feathering paddle wheel, multiply the quotient by .32. For a radial paddle wheel there should be one bucket for each foot of effective diameter; to find the number of buckets for a feathering paddle wheel, add 2 to the effective diameter, in feet, and divide the sum by 2.*

Or,

$$A = \frac{\text{I. H. P.} \times C}{D_e}$$

$$n = D_e \text{ for radial paddle wheels}$$

$$n = \frac{D_e + 2}{2} \text{ for feathering paddle wheels}$$

where A = area of one bucket, in square feet;
I. H. P = indicated horsepower;
C = a constant varying between .175 and .25 for radial wheels, and taken as .32 for feathering wheels;
D_e = effective diameter, in feet;
n = number of buckets.

If the side wheels are driven by separate engines, their combined horsepower is to be used. For a stern-wheel steamer, the area of the bucket may be about twice that given by the rule. For side-wheel steamers, the depth of the buckets is usually made about one-fourth their width.

EXAMPLE.—Find the area of each bucket and their number for a side-wheel steamer fitted with feathering paddle wheels 25 feet in effective diameter, each wheel being driven by a separate engine of 500 indicated horsepower.

SOLUTION.—Applying the rule given,

$$A = \frac{(500 + 500) \times .32}{25} = 12.8 \text{ sq. ft.}$$

and

$$n = \frac{25 + 2}{2} = 13.5 = 14. \text{ Ans.}$$

SCREW PROPELLERS

DEFINITIONS

27. If a point be caused to rotate at a uniform distance from and about an axis, and if the point at the same time be caused to advance at a uniform rate in the direction of axis, its' path will be a *helix*. If the point, when moving away from the observer, moves in the direction of the hands of a watch, the helix will be *right-handed;* if in an opposite direction, *left-handed*. The distance the point advances in one complete revolution is known as the *pitch*. If a line passing through the axis be caused to rotate about the axis, and to pass along the path of the point mentioned above, its path will be the surface of a *true screw*, provided the angle that the line makes with the axis remains constant. From this, it follows that a true screw is one in which the advance of any point, in the direction of the axis, at any distance from it, for any part of a revolution, but the same in each case, is the same. By causing lines making equal angles with each other and the axis to rotate about the axis in a helical path, a *multiple-threaded true screw* will be generated, having the same pitch as a single-threaded true screw generated by a line following the same helical path.

28. Consider a four-threaded, right-handed screw, generated by the lines OA, OB, OC, and OD, Fig. 3. These lines represent the intersections of the four helical surfaces with a plane EF perpendicular to the axis. Assume the helical surfaces to be cut by a plane, as GH, parallel to the first and intersecting the axis at another point. Then, $OAA'O'$, $OBB'O'$, $OCC'O'$, and $ODD'O'$ will be the helical surfaces of the blades of a four-bladed, right-handed screw propeller. If pieces of metal be shaped to conform to these helical surfaces and if these pieces of metal, which are called **blades**, be fastened to a hub, which in turn is keyed to a shaft rotated by an engine, the **screw propeller** in its simplest form is obtained. If the screw propeller is

revolving in the direction of the arrow, that portion of the blade that strikes the water first, which will be near the plane GH, is known as the **anterior portion** of the blade; and the portion that is near the plane EF, as the **posterior portion.** That part of the blade that is near the periphery AA' is known as the **tip.** In practice, screw propellers are hardly ever made of the shape shown in Fig. 3. Generally the anterior portion of the blade is rounded off toward the

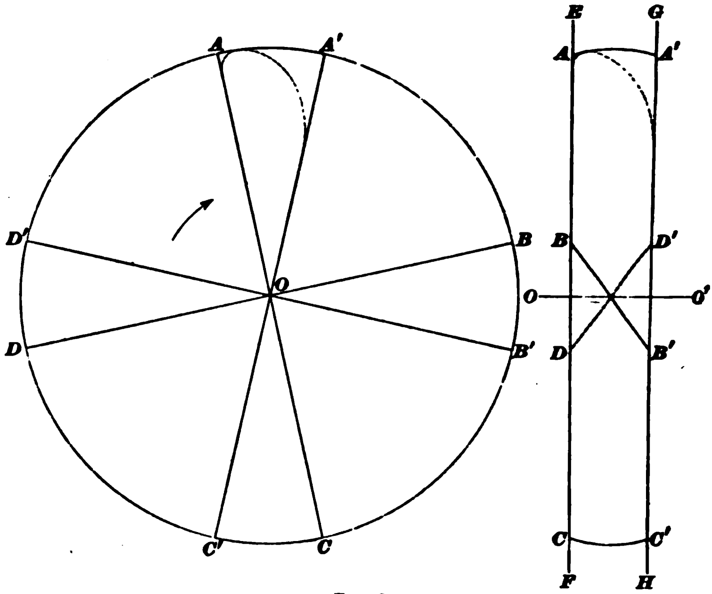


FIG. 3

tip, as shown by the dotted line on the blade $OAA'O'$. The posterior portion is also slightly rounded. Very often part of the anterior portion near the hub is also cut away.

29. Sometimes the surfaces of the blades are not truly helical; as usually found, the pitch near the tip is greater than the pitch near the hub. Such a propeller is said to have a *radially expanded pitch.* The reason for constructing the blade in this manner is this: Since the part of the blade near the hub strikes the water at nearly a right angle, it acts chiefly to churn the water, and since the water near the

periphery is thereby disturbed, the tip of the blade acts on water in motion. By increasing the pitch at the tip, it is supposed that the resistance at all parts of the blade is more nearly equalized.

The blades are sometimes constructed in such a manner that the anterior portion of the blade has a finer pitch than the posterior portion. Such a blade is said to have an *expanding* or *axially expanded pitch*. The object to be attained by it is as follows: The anterior portion of the blade, striking on water at rest and encountering the resistance due to a solid body moving through water at rest, sets the water in motion, driving it astern. Therefore, the posterior portion acts on water in motion. By expanding the pitch to the same extent, further motion is given to the water by the posterior portion, and it is supposed that the resistance at all parts of the blade is thereby equalized, the same as with radially expanded pitch blades.

From the explanations given it follows that, while radially expanded pitch blades are supposed to equalize the resistance at different parts of the blade at varying distances from the axis, expanding pitch blades are supposed to equalize the resistance at different parts of the blade at the same distance from the axis.

Neither radially expanded nor expanding pitch screw propellers seem to have met with the success claimed for them by their advocates, and, although at one time they obtained great favor, at present most vessels are fitted with propellers having blades forming helical surfaces.

30. The actual area of the surface on the driving side of a propeller blade is known by various names, as the *developed blade area*, the *helicoidal blade area*, or simply the *blade area*. When referring to the total blade area, it is usually spoken of as the *developed propeller area*, the *helicoidal propeller area*, or simply, the *propeller area*. The area of a blade projected on a plane at right angles to the propellor shaft is called its *projected area;* the projected area of all the blades is the *projected propeller area*. The area of the circle described by the

tips of the blades is the *disk area* of the propeller. The *pitch ratio* is the ratio of the pitch of the propeller to its diameter; it is usually expressed by giving its value; that is, the quotient obtained by dividing the pitch by the diameter.

MEASUREMENT OF PITCH

31. As stated in Art. **27**, the distance that a point moving in a helical path advances in the direction of the axis in one complete revolution is the pitch. Since the screw propeller blade, however, contains but a small portion of the line generated by a point moving in a helical path when the shaft makes one revolution, the pitch must be calculated by first finding the distance that a point, at a uniform distance

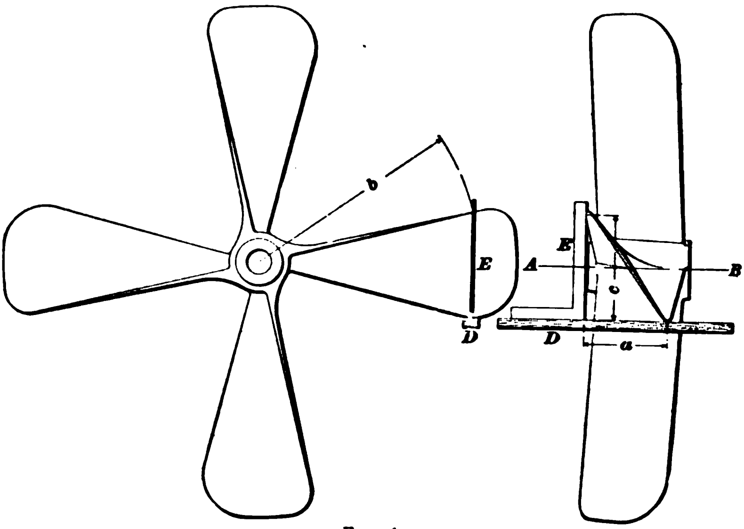


FIG. 4

from the axis, advances for that part of a helix that can be represented on the propeller blade; by the rules of proportion, the distance that a point would advance if the helical surface were continuous can then be found. In practice, the pitch of a propeller may be found quite closely in the manner illustrated in Fig. 4. Take a piece of joist or lath *D*, which

should be as straight as possible, and place it so as to touch one of the blades at any distance, as b, from the axis $A B$, taking care to hold it parallel to the axis. Next take a carpenter's square, shown at E, and place it on the lath and against the blade, so that the point at which the square touches the blade will be the same distance from the axis as is the lath. Measure the distances a, b, and c; a being the distance from the square to the point at which the lath touches the blade, and c the distance from the point at which the square touches the blade to the lath. The distances a and c may be obtained in a different manner, if considered more convenient, thus: Place the screw propeller so that one blade is horizontal. To a piece of string about 10 feet, or more, in length tie two nuts; place the string over the blade, with the nuts hanging down, at the distance from the shaft axis at which it is desired to find the pitch, taking care to so place the string that both parts hanging down are the same distance from the axis. The distance the two parts are apart is the distance a, Fig. 4. To find c, hold a lath against the blade and both vertical parts of the string; while holding the lath parallel to the shaft axis the distance c can be measured.

In a third method, which is a favorite with English marine engineers, the distance a is found by measuring with a lath from the after face of the stern post, or bracket, or using a string with two nuts. The width of the face of the blade, at the points the distance a was taken, is then measured, and the distance c is calculated by extracting the square root of the difference between the square of the width of the face of the blade and the square of the depth of the blade (the distance a, Fig. 4). Since, in the first and second methods given, the distance c is measured directly, these methods are preferable.

Having obtained the three measurements required, the pitch may be found very closely from the following proportion:

$$c : \text{circumference of circle having radius } b = a : \text{pitch}$$

whence, pitch $= \dfrac{2\pi b a}{c}$, where $2\pi = 2 \times 3.1416 = 6.2832$.

The directions given for finding the pitch may be stated in the form of a rule, thus:

Rule.—*To find the pitch of a given screw propeller, at a given distance from the center of the shaft, measure the depth of the blade at that distance and parallel to the shaft; measure the width of the blade at right angles to the shaft and at the same distance. Multiply the depth of the blade by the distance from the center of the shaft and by 6.2832; divide the product by the width of the blade. All dimensions are to be taken in inches.*

Or, $$P = \frac{6.2832\,b\,a}{c}$$

where P = pitch of screw propeller;
a = depth of blade;
b = distance from center of shaft where width and depth of blade is measured;
c = width of blade.

EXAMPLE.—If a screw propeller blade 6 feet from the center of the shaft is 22 inches deep and 41 inches in width, at right angles to the shaft, what is the pitch?

SOLUTION.—Applying the rule given,

$$P = \frac{22 \times 6 \times 12 \times 6.2832}{41} = 242.746 \text{ in.}$$

Or, $$\frac{242.746}{12} = 20 \text{ ft. } 2.746 \text{ in. Ans.}$$

32. In practice, it is advisable to take the measurements a and c, Fig. 4, at three or four distances from the axis, and calculate the pitch separately for each set of measurements. If the calculated pitches agree within a small percentage of error, the propeller will be either a true screw or one having expanding pitch blades. If the difference in the calculated pitches be considerable, it will show the propeller to be of radially expanded pitch; in that case, add the pitches together and divide by the number of them to obtain the mean or average pitch.

To find whether a propeller is constructed with expanding pitch or as a true screw, proceed as follows: Let Fig. 5 be a section through the blade at any convenient distance from the axis; the full lines show the outline of an expanding pitch blade, the dotted lines the outline of a blade forming part of a true screw.

Measure the distances *a* and *b* on the posterior portion, and the distances *c* and *d* on the anterior portion; also the distance from the axis at which the measurements are taken. Calculate the pitches for the anterior and posterior portions separately. If the pitches agree very closely, say within an error of 1 per cent., the blade is part of a true screw; if otherwise, the screw has an expanding pitch. If found to be the latter, add the two together and divide by 2 to get the mean pitch. If greater accuracy is required, the pitch may be calculated for any convenient number of portions of the blade, and the mean pitch found as above.

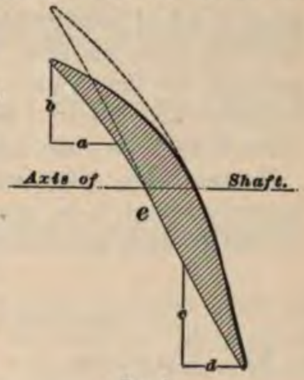


FIG. 5

The measurements for pitch should always be taken on the side of the blade that strikes the water when propelling the vessel ahead. In Fig. 5, which is the blade of a right-handed propeller, the surface *e* is the proper one on which to make the measurements.

The directions given may, in other words, be stated as follows:

To determine whether a screw propeller is a true screw, two or more measurements of the pitch should be taken on different parts of the blade at the same distance from the axis. Another set of measurements should be taken at some other distance from the axis. If the pitches calculated from these measurements agree closely, the propeller is a true screw.

To determine whether the pitch of the screw is radially expanded, calculate the pitch at two or more distances from the axis; if the pitch increases toward the tip of the blade, the screw propeller is of radially expanded pitch.

To determine whether the screw has an expanding pitch, the pitch must be calculated for the anterior and posterior portions of the blade. The pitch for the posterior portion should be the coarser; and, if calculated for any distance from the axis, the pitches of the anterior portion, as well

as those of the posterior portion of the blade, should agree, provided that the axial measurements are taken in the same planes passing through the axis.

33. Screw propellers, the blades of which form practically no screw at all, are sometimes found. To determine their pitch, a set of measurements should be taken at equidistant intervals from the axis, both for the anterior and posterior portions of the blade; the pitches calculated are added together, and the sum divided by the number of pitches in order to find the mean pitch.

In measuring a screw propeller for the pitch, it is well to remember that it is only necessary to measure one blade, no matter what the number of blades may be.

SLIP OF SCREW PROPELLER

34. The blades of a screw propeller drive a stream of water astern by their oblique action on the water when the instrument is revolved. The velocity of the stream thus projected is taken as that found by multiplying the pitch, in feet, by the number of revolutions in a given time. If the speed of the vessel in relation to the surrounding water clear of the wake be known, the apparent slip is calculated by rule II, Art. **17**. The true slip is calculated by rule I, Art. **17**. The apparent slip of a screw propeller will average about 10 per cent., except in freight steamers running at low speed and having a full under-water body, where the slip usually averages about 5 per cent.

SIZE OF SCREW PROPELLER

35. Pitch.—The pitch required for a screw propeller is found by the following rule:

Rule.—*To find the pitch of a proposed screw propeller, in feet, subtract the percentage of apparent slip, expressed decimally, from 1 and multiply the remainder by the number of revolutions per minute. Divide the desired speed of the vessel, in reference to the water clear of the wake, in feet per minute, by this product.*

Or,
$$P = \frac{V_s}{(1 - S_a)N}$$

where P = pitch, in feet;
V_s = speed of ship, in feet per minute, shown by log;
S_a = apparent slip, in per cent., expressed decimally;
N = revolutions per minute.

EXAMPLE.—Find the pitch for a vessel to make 12 knots* at 60 revolutions with 10 per cent. slip.

SOLUTION.— 12 knots $= \frac{12 \times 6{,}080}{60} = 1{,}216$ ft. per min. Applying the rule given,
$$P = \frac{1{,}216}{(1 - .1) \times 60} = 22.5 \text{ ft., nearly. Ans.}$$

36. Diameter.—For the diameter of a screw propeller, Seaton gives the following rule:

Rule.—*To find the diameter of a screw propeller, divide the indicated horsepower by the cube of the product of the pitch, in feet, and the revolutions per minute. Extract the square root of the quotient and multiply it by a constant ranging from 17,000 for slow freight steamers to 25,000 for fast-running light steamers, as torpedo boats and fast steam launches.*

Or,
$$D = C\sqrt{\frac{\text{I. H. P.}}{(PN)^3}}$$

where D = diameter of screw propeller, in feet;
I. H. P. = indicated horsepower;
P = pitch of screw propeller, in feet;
N = revolutions per minute;
C = a constant ranging between 17,000 and 25,000.

EXAMPLE.—Find the diameter of a screw propeller for a steam launch with an engine of 10 horsepower, the screw having a pitch of 4 feet and making 200 revolutions per minute.

SOLUTION.—Applying the rule given,
$$D = 25{,}000 \times \sqrt{\frac{10}{(4 \times 200)^3}} = 3.5 \text{ ft. Ans.}$$

When the rule gives a diameter that is impossible for the conditions, either P or N, or both, must be varied. Making either or both of these values larger will give a smaller

*A knot is 1 nautical mile per hour; it is not a distance.

diameter of screw; conversely, making either or both of these values smaller gives a larger diameter of screw. The rule is intended for screw propellers with four blades; if three blades are to be used, the diameter should be increased about 10 per cent.; and if two blades are to be used, about 20 per cent. The pitch ratio $\left(\frac{\text{pitch}}{\text{diameter}}\right)$ varies in practice between 1.1 and 1.6.

37. Blade Area.—The total actual blade area of screw propellers, according to Prof. W. F. Durand, for four-bladed screw propellers is made from 35 to 45 per cent. of the disk area. For a three-bladed screw, the total blade area varies between 27 and 33 per cent. of the disk area; and for two-bladed screws, between 20 and 25 per cent. of the disk area. The value to be chosen should vary with the pitch ratio, using a low total blade area for a low pitch ratio and increasing the value as the pitch ratio is made greater. The same authority states that ordinarily nothing is to be gained by making the total blade area more than 48 per cent. of the disk area.

38. Summary.—With the present imperfect knowledge of the action of screw propellers, no exact rules can be given for predetermining the pitch, diameter, and blade area, and also, no rules based on purely theoretical considerations that will at the same time conform with actual practice are possible at present. The rules given are empirical to a large extent; they are not intended to supersede good judgment based on actual experience with the action of screw propellers, but will furnish a guide tending to prevent judgment from going far astray.

EXAMPLES FOR PRACTICE

1. If a vessel is descending a river running 3 miles per hour, and at the expiration of 12 hours is 140 miles from the port of departure, what is: (*a*) its speed in relation to the port of departure? (*b*) its speed through the water? Express the speed in miles per hour.

Ans. $\begin{cases} (a) \text{ 11.67 mi. per hr.} \\ (b) \text{ 8.67 mi. per hr.} \end{cases}$

2. A screw propeller has a pitch of 20 feet and makes 70 revolutions per minute when it drives the ship at the rate of 12 knots; if the wake velocity is 2 knots, what is: (*a*) the true slip? (*b*) the apparent slip?

Ans. { (*a*) 27.62 per cent., nearly
(*b*) 13.14 per cent., nearly

3. Assuming the diameter of the rolling circle of a paddle wheel to be 18 feet, what is the percentage of apparent slip if its effective diameter is 23 feet? Ans. 21.74 per cent.

4. The diameter of the rolling circle being 20 feet, and the revolutions 20 per minute, what is the speed of the vessel, in miles per hour, counting 5,280 feet to the mile? Ans. 14.28 mi. per hr.

5. What should be the effective diameter of a stern wheel for a vessel to make 15 statute miles per hour at 25 revolutions per minute, allowing a slip of 20 per cent.? Ans. 21 ft.

6. What should be the pitch of a screw propeller to drive a ship 15 knots at 80 revolutions per minute with an apparent slip of 10 per cent.? Ans. 21.1 ft.

PROPULSION CALCULATIONS

THRUST

THEORETICAL, ACTUAL, AND INDICATED THRUST

39. A stream of water projected from a vessel propels the vessel by its reaction. According to Newton's third law of motion, there is always to every action an equal and opposite reaction. Hence, if the magnitude of the action is calculated, the magnitude of the reaction is known, since both are equal. The reaction of the stream of water projected from a vessel is known as the **thrust**, and is equal to the force required to project it. The magnitude of this force, and, hence, of the thrust, may be calculated by the following rule:

Rule.—*To find the magnitude of the theoretical thrust, multiply together the weight of the stream projected from the vessel, in pounds per second, and its velocity in regard to the*

surrounding water, in feet per second, that is, the true slip. Divide the product by 32.16.

Or,
$$T = \frac{WV}{g}$$

where W = weight of stream projected from vessel, in pounds per second;

V = its velocity in regard to surrounding water, in feet per second;

g = acceleration due to gravity, taken as 32.16;

T = theoretical thrust.

The rule just given is general; that is, it may be applied to any case, whether the vessel is propelled by paddle wheels, screw propeller, or a jet. The thrust found is the **theoretical thrust.** In practice, however, it cannot be calculated exactly, as only an approximation to the actual weight of the water projected in 1 second, and to its velocity in regard to the surrounding water, can be obtained. In thrust calculations, the weight of a cubic foot of sea-water is taken as 64.1 pounds, and the weight of a cubic foot of fresh water as 62.5 pounds.

The **actual thrust** is the real measure of the propelling force, and is always equal to the resistance of the vessel, in pounds. It cannot be calculated, but must be found by means of an instrument commonly known as a *dynamometer.*

EXAMPLE.—A paddle-wheel steamer, fitted with wheels having an effective diameter of 25 feet, makes 16 knots. If the revolutions per minute are 26, the width of the buckets 8 feet, and their depth 2 feet, what is the theoretical thrust in sea-water?

SOLUTION.—The velocity with which the paddle wheels project the stream is $\frac{26 \times 25 \times 3.1416}{60} = 34$ ft. per sec. The velocity of the vessel, in feet per second, taking the knot as 6,080 ft., is $\frac{16 \times 6{,}080}{60 \times 60} = 27$ ft. Then, the velocity of the stream, in relation to the surrounding water, is $34 - 27 = 7$ ft. The cross-sectional area of the stream is equal to the area of the buckets; that is, $8 \times 2 = 16$ sq. ft., and as there are two wheels, and, hence, two streams, the total area is $2 \times 16 = 32$ sq. ft. As the streams are projected at a velocity of 34 ft. per sec., their combined cubical contents are $32 \times 34 = 1{,}088$ cu. ft., and since the weight

of 1 cu. ft. of sea-water is 64.1 lb., the combined weight of the two streams is 1,088 × 64.1 = 69,740.8 lb. Applying the rule given,

$$T = \frac{69,740.8 \times 7}{32.16} = 15,179.9 \text{ lb. Ans.}$$

40. To find the weight of the stream projected by a screw propeller, the diameter and pitch of screw, the diameter of the hub, and the number of revolutions per second must be known. The following example shows how, from these data, the theoretical thrust may be calculated:

EXAMPLE.—A vessel, fitted with a screw propeller 14 feet in diameter and 18 feet pitch, makes 14 knots when the engine is making 90 revolutions a minute. The diameter of the hub is 3 feet. Find the theoretical thrust in sea-water, neglecting the wake velocity.

SOLUTION.—First, the cross-sectional area of the stream projected by the propeller must be found. This is assumed to be equal in area to the difference in areas between two circles having diameters equal to the screw and hub, respectively. The difference in area = $14^2 \times .7854 - 3^2 \times .7854 = 153.938 - 7.069 = 146.869$ sq. ft. The velocity of the stream, in relation to the vessel, is $\frac{18 \times 90}{60} = 27$ ft. per sec. The velocity of the vessel per sec. is $\frac{14 \times 6,080}{60 \times 60} = 23.64$ ft., nearly. Hence, the velocity of the stream, in regard to the surrounding water, is $27 - 23.64 = 3.36$ ft. a sec. The weight of the stream projected in 1 sec. is $27 \times 146.869 \times 64.1 = 254,186.18$ lb. Applying the rule given in Art. **39**,

$$T = \frac{254,186.18 \times 3.36}{32.16} = 26,556.77 \text{ lb. Ans.}$$

41. The term **indicated thrust** may be defined as the measure of the total force exerted by the propelling mechanism. It derives its name from the fact that it is calculated from the indicated horsepower of the engine. The indicated thrust should not be confounded with the actual thrust; the latter is the value of the net force actually propelling the vessel, while the former is the sum of the net force usefully applied to propelling the vessel and the force lost in overcoming all the frictional and other resistances.

A horsepower is 33,000 pounds raised 1 foot in 1 minute. The total work done by the engine, in foot-pounds per minute, is 33,000 × the indicated horsepower. Since work

is the product of force and distance, the magnitude of the force, whose reaction is the indicated thrust, may be found if the distance through which the force acts in 1 minute is known. This distance is found by multiplying the pitch of the screw propeller, in feet, by the number of revolutions per minute.

Rule.—*To find the indicated thrust of a screw propeller, divide 33,000 times the indicated horsepower of the engine by the product of the pitch, in feet, and the number of revolutions per minute.*

Or, $$T_i = \frac{33{,}000\,H}{P\,R}$$

where H = indicated horsepower of engine;
P = pitch of screw propeller, in feet;
R = revolutions per minute;
T_i = indicated thrust.

EXAMPLE.—Find the indicated thrust when the pitch is 22 feet, and the indicated horsepower developed by the engine, when making 94 revolutions, is 1,034.

SOLUTION.—Applying the rule given,

$$T_i = \frac{33{,}000 \times 1{,}034}{22 \times 94} = 16{,}500 \text{ lb. Ans.}$$

42. By reasoning similar to that from which the rule in Art. **41** is deduced, a rule for finding the indicated thrust of a paddle wheel may be obtained. The work done, as in the former case, is 33,000 × the indicated horsepower. The distance through which the force acts is equal to the speed, in feet per minute, of a point on the effective diameter circle.

Rule.—*To find the indicated thrust of a paddle-wheel steamer, divide 33,000 times the indicated horsepower of the engine by the product of 3.1416 times the effective diameter of the wheel, in feet, times the number of revolutions per minute.*

Or, $$T_i = \frac{33{,}000\,H}{3.1416\,D_e\,R}$$

where D_e is the effective diameter of the paddle wheel, in feet; the other letters denote the same values as in the formula given in Art. **41**.

EXAMPLE.—Find the indicated thrust when the effective diameter of paddle wheels is 26 feet and the indicated horsepower developed by the engine, at 31 revolutions, is 930.

SOLUTION.—Applying the rule given,

$$T_i = \frac{33{,}000 \times 930}{3.1416 \times 26 \times 31} = 12{,}120.2 \text{ lb. Ans.}$$

In applying the rule just given, it should be borne in mind that the total thrust is calculated. In case two wheels are driven by one engine, the thrust of each wheel is one-half the calculated thrust. In case of each wheel having a separate engine, as in some Western-river steamboats, the horsepower of the engine driving the wheel should be used to find the thrust of that wheel. If one wheel is driven by two engines, as in stern-wheel steamers, the combined horsepower of the two engines should be used.

THRUST BEARINGS

43. The thrust of the paddle wheels is taken by the bearings in which the paddle-wheel shaft is supported. The outboard bearings are usually bolted to a bracket securely attached to the side of the vessel; against these bearings, the whole of the thrust is exerted. The bearings may be

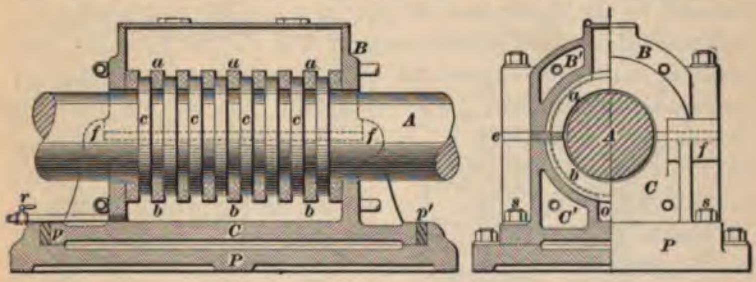


FIG. 6

of any suitable form, provided that they are strong enough to withstand the thrust.

With a screw propeller, a special form of bearing is necessary. The simplest form of such a bearing is shown in Fig. 6. The thrust of the screw propeller tends to force the propeller shaft either inboards or outboards, according

to the direction in which the vessel is propelled. To resist this tendency, a number of collars *c* are forged on the thrust shaft *A*; these collars bear against semicircular bronze thrust rings *a*, *b*, placed within recesses formed in the cap *B* and base *C*, respectively. To prevent any rotation of the thrust rings, a tongue *e* is placed between the cap and base at each side of the shaft. To prevent any longitudinal movement of the cap, it is fitted carefully between lugs *f*, *f* cast solid with the base; it is also bolted to the latter. Both the cap and base are cored out, as shown at *B'* and *C'*; they are usually fitted with inlet and outlet pipes, by means of which water may be circulated through the cored chambers in order to prevent or alleviate any heating of the thrust rings that may occur while running. The whole arrangement, which is called a **thrust block,** is bolted to a pedestal *P* that is securely attached to the framing of the vessel. The thrust block may be adjusted to a limited extent in an axial direction by inserting a thin liner at *p* and removing one of the same thickness at *p'*, or vice versa. To allow such an adjustment to be made, the holes in the base through which the holding-down bolts *s*, *s* pass are made oblong. To provide for a constant lubrication of the collars, a reservoir *o* is formed in the lower part of the base. This is filled with a mixture of oil and soapy water. The collars dip into this mixture and carry it around with them, thus providing an automatic lubrication. The reservoir may be emptied by means of the petcock *r*. It is of the greatest importance that the thrust should be evenly distributed over all the collars and thrust rings; for, assuming the whole thrust to be exerted against but one or two rings, the pressure would be so great as to prevent lubrication. In the thrust block shown in Fig. 6, the thrust rings cannot be adjusted very readily; therefore, this style, which is known as a *solid thrust block*, is gradually going out of use.

44. The *sectional*, or *horseshoe*, *thrust block*, shown in Fig. 7, is taking the place of the solid thrust block. In this, each thrust ring may be adjusted very readily to a nicety,

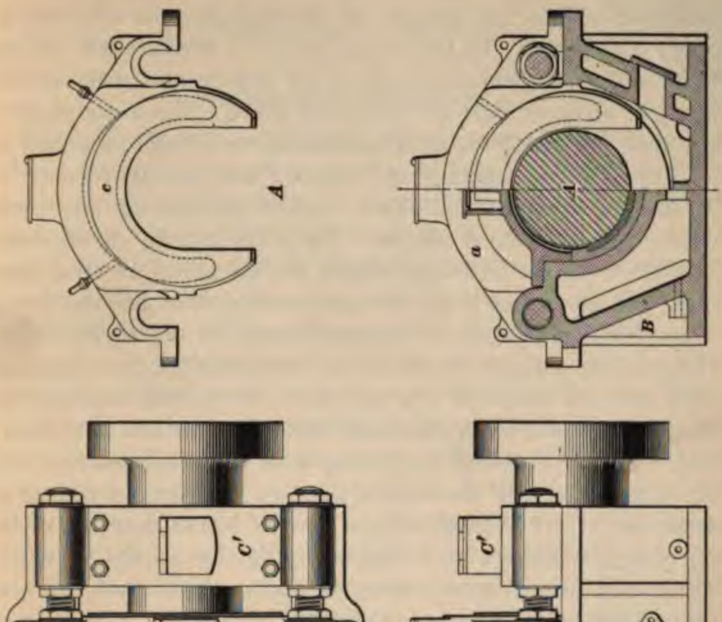


FIG. 7

independently of the others, or all rings may be adjusted at once, if required. In the figure, *A* is the thrust shaft; *B*, the base; and *C*, *C'*, the bearings placed at each end of the thrust block in order to support the thrust shaft. The thrust rings *a* are made in the form of a horseshoe, as shown in detail at *A*, Fig. 7. The wearing surfaces *c*, *c'* are usually formed of bronze for wrought-iron shafts, and of Babbitt, or any other white metal, for steel shafts. They are usually made separate from the thrust rings, which, in this case, are steel castings, and are either hung on steady pins driven into the thrust rings, or inserted into recesses formed in the thrust rings. Large screws *E*, *E'* are placed on each side of the block and pass through holes in the bearings *C*, *C'*; they are secured by nuts situated on the forward and after sides of each bearing. The thrust rings are placed over these screws and may be adjusted as well as locked, when in position, by means of nuts, as *e*, *e'*. The thrust rings, as well as the base, are cored out and provided with suitable piping, by means of which water may be circulated through them. The lower part of the base forms a reservoir for the lubricant.

45. The thrust block is usually located close to the engine, it being the common practice to make the thrust shaft the first length of shafting abaft of the engine. Thrust blocks of the horseshoe type are frequently bolted directly to the bedplate of the engine. If located there, they can be given the attention their importance demands. If heating and subsequent cutting of the thrust rings occur, which entail a very rapid wearing away of the rings, the whole line of shafting is thrown forwards an amount equal to their wear. Should the crank-shaft and line-shaft journals be grooved to any extent, their heating and subsequent cutting are very liable to occur as soon as the thrust block heats. Since the shafting and bearing may be seriously damaged by means of this, it is of great importance that all possible attention be given to the thrust block.

SPEED OF VESSELS

POWERING OF VESSELS

46. The exact amount of power required to propel a vessel at a given speed cannot be deduced very readily from the elementary principles of mechanics. Instead, empirical rules based on the actual performance of vessels are usually relied on. The conditions that influence the relation between power and speed are many, but only a few of the more important ones will be enumerated here. For instance, the area of the blades of the screw propeller may not be sufficient for high speed, owing to a churning of the water when the propeller is revolved beyond a certain speed; and, although the power expended in revolving the propeller faster may be considerable, the increase of the speed of the vessel may be very slight. A similar state of affairs may occur if the area of the buckets of a paddle wheel is too small. It may be amply sufficient for a low rate of speed, and still be entirely too small for a higher rate, thus showing, probably, a high efficiency of the propelling instrument at a low speed, and a very poor one at a higher rate. Again, the efficiency of the engine may vary greatly for different powers developed by the same engine. Therefore, no hard and fast rule can be laid down that will express the relation between power and speed under all conditions.

47. The rule most frequently used in the powering of vessels is known as the *Admiralty rule.* It involves the selection of a proper constant based on actual experience; when this constant, a number of which are given in Table I, due to Mr. Seaton, is properly selected, the results of the rule will be found to agree very closely with the actual performance of vessels powered by the rule, at least under ordinary conditions and for ordinary efficiencies of the propelling apparatus.

Rule.—*To find the indicated horsepower required to propel a vessel at a given speed, multiply together the cube of the speed*

and the cube root of the square of the displacement. Divide this product by the constant corresponding to the length, speed, and shape of the vessel, as given in Table I.

Or, $$H = \frac{S^3 \sqrt[3]{W^2}}{k}$$

where H = indicated horsepower;
W = displacement of vessel, in tons of 2,240 pounds;
k = a constant (see Table I);
S = speed, in knots.

TABLE I

VALUES OF k IN ADMIRALTY RULE

Description of Vessel	Speed Knots	k
Under 200 feet, fair	9 to 10	200
Under 200 feet, fine	9 to 10	230
Under 200 feet, fine	10 to 11	210
Under 200 feet, fine	11 to 12	200
From 200 to 250 feet, fair	9 to 11	220
From 200 to 250 feet, fine	9 to 11	240
From 200 to 250 feet, fine	11 to 12	220
From 250 to 300 feet, fair	9 to 11	250
From 250 to 300 feet, fair	11 to 13	220
From 250 to 300 feet, fine	9 to 11	260
From 250 to 300 feet, fine	11 to 13	240
From 250 to 300 feet, fine	13 to 15	200
From 300 to 400 feet, fair	9 to 11	260
From 300 to 400 feet, fair	11 to 13	240
From 300 to 400 feet, fine	11 to 13	260
From 300 to 400 feet, fine	13 to 15	240
From 300 to 400 feet, fine	15 to 17	190
Above 400 feet, fine	15 to 17	240

48. To determine whether a vessel is fair or fine, it is usual to compare its displacement, in cubic feet, with the volume of a rectangular box having a length equal to the

length of the vessel on the water-line, a width equal to the beam, and a depth equal to the mean draft of the vessel diminished by the depth of the keel. If the displacement is .55 of the volume of the box, or less, the vessel is fine; if above .55 and less than .70, fair. The quotient obtained by dividing the displacement by the contents of the imaginary box is called the *coefficient of fineness*.

EXAMPLE.—A vessel, 260 feet long and finely shaped, having a displacement of 1,000 tons, is to have a speed of 15 knots; what should be the indicated horsepower of the engine?

SOLUTION.—From the table, $k = 200$. Applying the rule given,

$$H = \frac{15^3 \times \sqrt[3]{1{,}000^2}}{200} = 1{,}687.5 \text{ I. H. P.} \quad \text{Ans.}$$

49. The selection of a proper value of k calls for the exercise of considerable judgment, based on personal knowledge of the actual performance of similar vessels. Generally speaking, the value of k is influenced by the length, speed, and shape of the vessel. The value of k should be greater with an increased length of the vessel in proportion to the width, and also with a finer under-water body; conversely, its value should be less as the ratio of length to width becomes smaller, and as the form becomes fuller. Furthermore, the value of k should be smaller for relatively high speeds than for low speeds, for vessels of the same form and displacement. Prof. W. F. Durand states that a speed may be considered as relatively high or low if the speed exceeds the numerical value of the square root of the length of the vessel in feet, or falls below it. Thus, if a vessel is 64 feet long, $\sqrt{64} = 8$, a speed of 10 knots would be considered as relatively high, while a speed of 5 knots would be considered as relatively low. For small boats, if the speed is given in statute miles per hour, the values of k range between 150 and 225, and for speeds given in knots, between 100 and 150, according to Prof. W. F. Durand, in an article contributed to "Marine Engineering."

EXAMPLE 1.—A boat 70 feet long and 7 feet beam is to make 10 knots; its displacement is 30 tons, and the vessel has fair lines, having a coefficient of fineness of .65. What horsepower is required?

SOLUTION.—As the speed is relatively high, it will be well to select a rather low value of k, say 120. Applying the rule given in Art. 47,

$$H = \frac{10^3 \times \sqrt[3]{30^2}}{120} = 80.46, \text{ say } 81, \text{ I. H. P. Ans.}$$

EXAMPLE 2.—A boat 60 feet long and 8 feet beam is to make 9 knots; its displacement is 30 tons, and it has fair lines. What horsepower is required?

SOLUTION.—As the speed is relatively high, a low value of k should be selected on this account. Furthermore, as the ratio of length to width is small, the value of k should be decreased on this account also. Selecting a value of 110, by the rule given in Art. 47,

$$H = \frac{9^3 \times \sqrt[3]{30^2}}{110} = 64 \text{ I. H. P., nearly. Ans.}$$

EXAMPLE 3.—A boat 50 feet long and 7 feet beam is to make 6 knots; its displacement is 25 tons and it has fine lines. What horsepower is required?

SOLUTION.—The speed being relatively low, and the lines fine, a rather high value of k, say 140, can be selected. Applying the rule given in Art. 47,

$$H = \frac{6^3 \times \sqrt[3]{25^2}}{140} = 13.2 \text{ I. H. P., nearly. Ans.}$$

ENGINE SPEED AND SHIP'S SPEED

50. Very often, the question as to the speed at which the engine must be run to drive the vessel at a certain velocity, confronts the marine engineer. If the revolutions per minute of the engine for a certain speed of the vessel are known, the question may be readily answered. Assuming the percentage of slip to remain constant, doubling the velocity of the stream projected by the propelling instrument, that is, doubling the revolutions of the engine, and, hence, of the screw propeller or paddle wheels, doubles the speed of the vessel; in other words, the speed is directly proportional to the revolutions of the engine. In actual practice, the percentage of slip varies somewhat at different speeds and under different conditions; hence, the following rule, which is based on the assumption of a constant percentage of slip, does not give the exact number of revolutions per minute required, which can be found only by actual trial. However, the rule will give a very fair approximation.

Rule.—*To find the revolutions per minute at which to run the engine in order to give the required speed, divide the product of the revolutions producing any given speed and the required speed by the given speed.*

Or,
$$R_1 = \frac{R\,S_1}{S}$$

where R = revolutions per minute for a given speed;
S = given speed;
R_1 = required revolutions;
S_1 = required speed.

EXAMPLE.—If a vessel is propelled at the rate of 16 knots when the engine is making 32 revolutions per minute, what should be the revolutions per minute to give it a speed of 14 knots?

SOLUTION.—Applying the rule given,
$$R_1 = \frac{32 \times 14}{16} = 28 \text{ rev. per min. Ans.}$$

51. On taking charge of a steamer, the number of revolutions at which to run the propelling instrument to produce a given speed, when none of the data called for in the rule given in Art. **50** are available, is often desired. In that case, the pitch of the screw, or the effective diameter of the paddle wheel (taking the effective diameter for this purpose from center to center of buckets) must be measured and a fair slip value assumed. The revolutions can then be found by the following rule:

Rule.—*To find the revolutions per minute required to produce a given speed per hour, multiply the pitch of a screw propeller, or the circumference of the effective diameter circle of a paddle wheel, by 60 and by the difference between 1 and the assumed apparent slip, expressed decimally. Divide the speed of the ship in feet per hour through the water by this product.*

Or,
$$R = \frac{S}{60\,D\,(1 - S_a)}$$

where R = revolutions per minute;
S = speed, in feet per hour;
D = pitch of screw propeller, or effective diameter $\times$ 3.1416 in case of a paddle wheel;
S_a = apparent slip, expressed decimally in per cent.

EXAMPLE.—The pitch of a screw propeller is 16 feet; how many revolutions per minute must it make to drive the ship at the rate of 10 knots, the apparent slip being estimated at 10 per cent.?

SOLUTION.—Applying the rule given,

$$R = \frac{6,080 \times 10}{60 \times 16 \times (1 - .1)} = 70.37 \text{ rev. per min. Ans.}$$

52. The probable speed of a ship with a given number of revolutions of the propelling instrument can be found as follows:

Rule.—*To find the probable speed of a steam vessel, in miles per hour, multiply 60 by the revolutions per minute and the pitch, in feet, of a screw propeller, or the circumference of the effective diameter circle of a paddle wheel, and by the difference between 1 and the per cent. of apparent slip expressed decimally. Divide the product by 5,280 or by 6,080, according to whether the speed is to be expressed in statute miles or nautical miles.*

Or,
$$S = \frac{60\,R\,D\,(1 - S_a)}{f}$$

where f is the number of feet per statute or nautical mile; the other letters have the same meaning as in the formula given in Art. **51.**

EXAMPLE.—A propeller of 20 feet pitch makes 70 revolutions per minute; with an assumed slip of 12 per cent., what will be the speed of the ship in knots?

SOLUTION.—Applying the rule just given,

$$S = \frac{60 \times 20 \times 70 \times (1 - .12)}{6,080} = 12.16 \text{ knots. Ans.}$$

RELATION BETWEEN HORSEPOWER AND REVOLUTIONS

53. The speed of a ship fully under way is about directly proportional to the number of revolutions made by the engine. But the power required to turn the propelling instrument varies as the cube of the number of revolutions, or (which is the same thing) as the cube of the speed. Assume some body (the shape of it is immaterial) to be moved through water at a uniform speed of 1,100 feet per minute, and assume that a constant propelling force of 16,500 pounds is required to maintain that rate of speed.

Since work is the product of force into distance, the work done per minute is $1,100 \times 16,500 = 18,150,000$ foot-pounds. Since a horsepower is 33,000 foot-pounds of work per minute, the horsepower required to move the body at the given speed is $\frac{18150000}{33000} = 550$ horsepower. Suppose that the same body is moved through the water at the uniform speed of 2,200 feet per minute, that is, twice as fast as before. Now, the resistance to a body moving through water varies as the square of the velocity; hence, the force required to move the body at double the speed will be $2^2 \times 16,500 = 66,000$ pounds. But, since the body will move the same distance as in the first case considered, i. e., 1,100 feet in one-half the time, or 30 seconds, the power required will be $\frac{66,000 \times 1,100 \times 60}{33,000 \times 30} = 4,400$ horsepower.

This shows that to double the speed the power had to be increased $2 \times 2 \times 2 = 8$ times; that is, it varies as the cube of the speed. Had the calculation been made for any other speed, say 1.5 times the original speed, the power required would have been found to be $1.5 \times 1.5 \times 1.5 = 3.375$ times as much. Likewise, for a speed treble that first considered, the power would have been $3 \times 3 \times 3 = 27$ times as much. From this the following rule is derived:

Rule.—*To find, approximately, the power developed by an engine for any given number of revolutions per minute, any other horsepower and the corresponding revolutions per minute being known, multiply together the cube of the revolutions at the required horsepower and the given horsepower. Divide the product by the cube of the revolutions corresponding to the given power.*

Or,
$$H_1 = \frac{R_1^3 H}{R^3}$$

where H_1 = required indicated horsepower;
R_1 = revolutions at required power;
H = given indicated horsepower;
R = revolutions corresponding to it.

EXAMPLE.—A tug engine develops 70 indicated horsepower at 100 revolutions; what power would it probably develop at 60 revolutions?

SOLUTION.—Applying the rule given,

$$H_i = \frac{60^3 \times 70}{100^3} = 15.12 \text{ I. H. P.} \quad \text{Ans.}$$

In practice, the horsepower calculated by this rule will not always correspond to that actually used, as found by the indicator diagram. This is due to the fact that the efficiency of the machinery is not necessarily the same at all speeds. As a general rule, the engine will be at its maximum efficiency at some certain speed, and will have a lower efficiency at a higher or lower speed. The speed at which the engine is at its best efficiency can be found only by actual trial.

REDUCTION OF HORSEPOWER WHEN TOWING

54. It is a well-known fact among marine engineers that an engine will develop a lower horsepower with a given boiler pressure, throttle position, and cut-off when towing or having the resistance of the vessel increased by other means, as by a head-wind, a head-sea, or an adverse current, than will be developed under the same engine conditions but running free. The reason for this is explained in the following discussion, in which for the sake of simplicity two convenient assumptions have been made that are not absolutely correct in practice. These assumptions are that the horsepower of an engine varies directly as the first power of the number of revolutions, the mean effective pressure remaining the same, and that the speed of the vessel varies directly as the first power of the number of revolutions.

Consider a paddle-wheel steamer running free, with its engine developing its greatest horsepower possible. Since the turning effort of the engine depends only on the mean effective pressure in the cylinders, it is independent of the revolution so long as the throttle position, boiler pressure, and cut-off remain the same. This turning effort, when exerted at the circumference of the effective diameter circle tangentially to the same and parallel to the surface of the water, is the total force tending to propel the vessel forwards, and is resisted by an opposing force, which is the resistance of the vessel. Let the engine be started and assume that it is

making its greatest turning effort. The resistance being less than the forward force, the vessel moves forwards under the influence of a forward accelerating force equal to the difference between the total forward force and the resistance. As the vessel gathers headway, the resistance increases; this means that the difference between the forward and the resisting force, that is, the accelerating force, decreases until the total forward force and total resistance have become equal, when the vessel continues at a uniform speed. Let the resistance be increased either by the vessel picking up a tow, by a head-wind or head-sea, by encountering an adverse current, or by a combination of these circumstances. The conditions remaining the same as before at the engine, the turning effort, that is, the total forward force is the same; but, as the initial resistance is increased, the initial difference between the total forward force and the total resistance is smaller than in the first case. This means that a smaller accelerating force is available with an increased resistance, and consequently the total forward force and total resistance become equal at a lower speed of the vessel, which then continues under way at a uniform, but lower speed. Now, the horsepower of an engine varies (theoretically) directly as the first power of the number of revolutions, the mean effective pressure remaining constant. It has been shown that the speed of the vessel has been lowered; from this it follows that the revolutions and consequently the horsepower must be less when the resistance has been increased. By adding to the resistance of the vessel, a condition is finally reached similar to that of a vessel moored to a dock; the forward force and resistance are equal and the vessel, as no accelerating force is available, remains stationary. In this condition, the number of revolutions, and, hence, the horsepower, has dropped to the lowest limit.

RELATION OF COAL CONSUMPTION TO SPEED

55. The fuel consumption may be said to vary directly as the horsepower developed (this is not exactly true, but only approximately). The horsepower varies about directly

as the cube of the speed, whence it follows that the fuel consumption will also vary as the cube of the speed (approximately). From this, the following rules have been deduced.

Rule I.—*To find the probable coal consumption for a speed different from a known speed, multiply the cube of the new speed, in miles per hour or knots, by the coal consumption, in tons, at the known speed. Divide the product by the cube of the known speed, expressed in the same terms as the new speed; the quotient will be the coal consumption expressed in the same terms as the known one.*

Or, $$c = \frac{s^3 C}{S^3} \qquad (1)$$

where c = coal consumption, in tons, at the new speed;
s = new speed;
S = known speed;
C = coal consumption, in tons, at the known speed.

EXAMPLE 1.—A steamer consumes 80 tons of coal per day at a speed of 12 knots; suppose that the speed is to be reduced to 10 knots, what will be the fuel consumption per day at that rate of speed?

SOLUTION.—Applying rule I,

$$c = \frac{10^3 \times 80}{12^3} = 46.3 \text{ T. per da., nearly. Ans.}$$

Rule II.—*To find approximately the speed of steaming for a new coal consumption, multiply the new coal consumption by the cube of the known speed, divide the product by the coal consumption at the known speed, and extract the cube root of the quotient. The coal consumption is to be expressed in the same terms in both cases, and the new speed will be in the same terms as the old speed.*

Or, $$s = \sqrt[3]{\frac{c S^3}{C}} \qquad (2)$$

where the letters have the same meaning as in formula **1**.

EXAMPLE 2.—A steamer consumes 100 tons of coal per day at a speed of 10 knots; what should be the speed in order to cut the coal consumption down to 50 tons per day?

SOLUTION.—Applying rule II,

$$s = \sqrt[3]{\frac{50 \times 10^3}{100}} = 7.9, \text{ or 8 knots, nearly. Ans.}$$

EXAMPLE 3.—If a steamer consumes 15 tons of coal per day to produce a speed of 9 knots, how many knots would she steam if the coal consumption were reduced to 12 tons per day?

SOLUTION.—Applying rule II,

$$s = \sqrt[3]{\frac{12 \times 9^3}{15}} = 8.35 \text{ knots, nearly. Ans.}$$

56. At sea, owing to an accident, it often occurs that it is desired to know what speed to maintain in order to reach a given port with the amount of coal on hand. This problem is readily solved by trial and by application of rule I, Art. **55.** In practice, a good margin of coal should be shown by the calculations as left over, for the reason that the actual coal consumption at the reduced speed will, as a general rule, be in excess of the calculated consumption, by reason of the decrease in economy of the engine induced by reducing the developed horsepower.

EXAMPLE.—A steamer consumes 20 tons of coal per day at a normal speed of 10 knots; the distance to the nearest port where coal can be had is 600 miles, and the estimated quantity of coal in the bunkers is but 35 tons. Find what speed should be maintained in order to reach the coaling station with the coal supply on hand.

SOLUTION.—The best way to proceed in a case of this kind is to assume a lower speed, say 8 knots, and calculate the new coal consumption for that speed; thus, $c = \frac{8^3 \times 20}{10^3} = 10.24$ T. per da., or .43 T. per hr. The time required to cover a distance of 600 mi. at a speed of 8 knots is $\frac{600}{8} = 75$ hr., and at a coal consumption of .43 T. per hr. the total quantity of coal required at that speed is $75 \times .43 = 32.25$ T. Hence, if a speed of 8 knots is maintained, the supply of coal on hand (35 T.) will suffice to reach the coaling station under ordinary weather conditions. Ans.

EXAMPLES FOR PRACTICE

1. If a stream of water weighing 32,160 pounds be projected per second from a vessel, what will be the theoretical thrust, assuming the true slip to be 2 feet per second? Ans. 2,000 lb.

2. Find the indicated thrust when the pitch of screw propeller is 20 feet; revolutions per minute, 60; indicated horsepower developed, 1,000. Ans. 27,500 lb.

3. The effective diameter of a stern wheel is 15 feet; what is the indicated thrust corresponding to 100 indicated horsepower at 30 revolutions?
Ans. 2,334.27 lb.

4. Find the indicated horsepower for a vessel 190 feet long, having a displacement of 1,000 tons, a coefficient of fineness of .53, and a speed of 12 knots.
Ans. 864 I. H. P.

5. At what number of revolutions should a screw propeller turn to drive a vessel at 15 knots, if at 60 revolutions it made 12 knots?
Ans. 75 rev.

6. If a screw propeller has a pitch of 14 feet, and its apparent slip is known from a trial-trip record to be 15 per cent., how many revolutions per minute must it make to drive the ship 12 statute miles per hour against a current running 2 miles per hour?
Ans. 103.5 rev., nearly

7. About what speed may be expected from a stern-wheel steamer if the wheel has an effective diameter of 17 feet and makes 32 revolutions per minute? The speed is desired in knots, and the apparent slip is estimated at 20 per cent.
Ans. 13.5 knots, nearly

8. If a marine engine develops 100 indicated horsepower at 90 revolutions, what will it develop at 100 revolutions?
Ans. 137.17 I. H. P.

9. A steamer consumes 20 tons of coal per day at a speed of 7 knots; what is the probable coal consumption at 11 knots?
Ans. 77.6 T. per da.

10. At what speed must a steamer proceed to cut the coal consumption down to 40 tons per day when at 60 tons per day it makes 18 knots?
Ans. 15.7 knots, nearly

REFRIGERATION

REFRIGERATING MACHINERY

FUNDAMENTAL PRINCIPLES

INTRODUCTION

1. Processes of Producing Cold.—The act of lowering the temperature of a body, or of keeping its temperature below that of the atmosphere, is spoken of as **refrigeration.** It may be produced by one of the following processes:

1. A transfer of heat from a warmer to a colder body.
2. A chemical action, as exemplified by the so-called freezing mixtures.
3. The adiabatic expansion of a gas.
4. The evaporation of liquids having a low boiling point.

Heat will pass from a warmer to a colder body. Drop a hot piece of iron into a vessel of cold water; the water will absorb heat from the iron until both the iron and the water are at the same temperature; that is, the heat of the warm body (the hot piece of iron) passes to the cold body (the cool water).

In actual practice, ice-making and refrigerating machines employ either of the last two processes in combination with the first. The second process has no practical applicability at present in commercial work.

In order to change a liquid into a vapor, a certain quantity of heat, called the *latent heat of vaporization*, must be added

to the liquid. When this process of vaporization or evaporation is taking place in the presence of other warmer bodies, the heat required for it is drawn from these bodies, and they are thereby cooled. From this, it follows that, with a liquid having the boiling point below the freezing point of water, water can be frozen by allowing the liquid to absorb the heat contained in the water, and thus acquire the heat required for its own evaporation.

2. Adiabatic and Isothermal Expansion and Compression.—Let a given volume of any gas be confined under pressure in a cylinder, like that of a steam engine; the gas will then tend to move the piston, that is, it will tend to overcome resistance. If the pressure is sufficient to overcome the resistance, the piston will move and the gas, in expanding, will be doing work. Now, if a thermometer is inserted in the cylinder, it will be found that as the gas expands its temperature is lowered. As is well known, it is not the steam itself that does work in a steam engine, but it is the heat contained in the steam. This statement applies to all other gases as well. Then, as heat must be given up in order to do work, it follows that if no heat is supplied from outside sources, a gas, in expanding, cannot do work without its temperature being lowered. The expansion of a gas not accompanied by a transfer of heat from another body, is known as **adiabatic expansion.** The word "adiabatic" is derived from the Greek word *adiabatos* (*a*, not; *diabainein*, to pass through), and is descriptive of a process in which there is no transfer of heat from or to a body operated on to or from another body. Conversely, if a given volume of a gas be compressed, its temperature is raised if no heat is abstracted from it during compression. That the temperature must become higher can be seen when it is considered that it is impossible to compress the gas into a smaller volume without doing work. The process of compression changes this work into heat; that is, heat is added to the gas, and consequently its temperature is increased. The compression of a gas, accompanied by an increase of temperature

proportional to the conversion of the work done in compressing it into heat, is known as **adiabatic compression.** In other words, a gas is said to expand or to be compressed **adiabatically** when no heat is added to it from an outside source during expansion or abstracted by any medium while the gas is compressed.

3. Let a given volume of gas under pressure expand and do work. As previously explained, it cannot do work without parting with an equivalent amount of heat. Let this same amount of heat be added from some outside source. Then, the gas, in expanding and doing work, will remain at a constant temperature, and it is now said to expand *isothermally*. The term "isothermal," which is derived from the Greek (*isos*, equal; *therme*, heat), simply means at a constant temperature. Conversely, if a given volume of gas at a given temperature be compressed, and the amount of work converted into heat in compressing it be abstracted by some means, its temperature will remain the same. This process is known as **isothermal compression.** In other words, a gas is said to expand or to be compressed **isothermally** when heat is added during expansion or abstracted during compression, to keep the temperature constant.

4. Suppose that a certain volume of gas at a given pressure is allowed to expand without doing any work. Then, assuming the vessel in which expansion is taking place to be a perfect non-conductor of heat, so that no heat can be added or abstracted by any outside means, the temperature of the gas during expansion will remain constant; that is, the expansion is adiabatic and isothermal at the same time.

5. The third method mentioned in Art. 1 suggests a mechanical process of refrigeration. Let a certain volume of any gas be confined in a cylinder and let it be compressed adiabatically by doing work on it. When compressed to the smallest volume feasible, reduce the temperature by allowing the heat due to the conversion of work into heat during compression to pass into a colder body, say cool

water. The gas having been cooled to the original temperature, or nearly to it, can be made to do work, expanding adiabatically. But in doing work, the pressure and temperature of the given quantity of the gas fall rapidly, and the temperature soon falls far below that of the atmosphere surrounding the machine in which expansion is taking place. Then, as heat will readily pass from a warmer to a colder body, the expanded gas, which is very cold, will, on being brought near a warmer body, absorb some of its heat, that is, cool it.

CAPACITY OF REFRIGERATING MACHINES

6. The **capacity** of a refrigerating machine is the measure of its ability to abstract heat. The unit of **refrigerating** or **ice-melting capacity** is the quantity of heat required to melt 1 ton (2,000 pounds) of ice at 32° F. to water at 32° F. The latent heat of fusion of water being 144 British thermal units, the unit of refrigerating capacity is equal to $144 \times 2{,}000 = 288{,}000$ British thermal units.

It is the practice of some British writers to use the long ton (2,240 pounds); on this basis, the unit of refrigerating capacity is equal to $144 \times 2{,}240 = 322{,}560$ British thermal units. This value is virtually obsolete at present, however.

Rule.—*To find the refrigerating capacity of an ice machine, divide the number of British thermal units abstracted in 24 hours by 288,000.*

Or,
$$F = \frac{H}{288{,}000}$$

where F = refrigerating capacity, in tons;
H = number of British thermal units abstracted per day of 24 hours.

EXAMPLE.—A refrigerating machine abstracts 2,375,241 British thermal units in 20 hours; what is its refrigerating capacity?

SOLUTION.—The heat units abstracted in a day are $\frac{2{,}375{,}241 \times 24}{20}$ B. T. U. Applying the rule just given,

$$F = \frac{2{,}375{,}241 \times 24}{20 \times 288{,}000} = 9.897, \text{ say 10 T. Ans.}$$

7. The ice-making capacity is the number of tons of ice that a machine is capable of producing in 24 hours. As the temperature of the water from which the machine makes the ice varies from 50° to 95°, and as it is necessary to cool this water to 32° before any ice can be made, it will be seen that the ice-making capacity is variable and is largely affected by the conditions under which the machine operates. Owing to the necessity of cooling the water from which the ice is made from its initial temperature to a temperature below the freezing point, and owing to other losses, such as radiation, etc., the ice-making capacity is only about 50 or 60 per cent. of the ice-melting capacity.

8. A refrigerating machine is generally driven by a steam engine; therefore, the energy delivered to the machine is contained primarily in the fuel fed to the furnace, usually coal. For this reason, it is customary in commercial work to measure the commercial efficiency, or the economy of a refrigerating machine, by the pounds of ice-melting effect per pound of coal used. For every pound of coal consumed in the boiler to produce steam to operate the refrigerating machine, a quantity of heat is abstracted from the cold body sufficient to melt a definite number of pounds of ice at 32° F. into water at 32° F. This quantity of ice is a measure of the commercial efficiency of the machine.

EXAMPLE.—A refrigerating machine having an actual ice-melting capacity of 23.5 tons requires 4,350 pounds of coal per 24 hours to operate it; what is the efficiency, expressed in ice, per pound of coal?

SOLUTION.— 23.5 T = 47,000 lb.; 47,000 ÷ 4,350 = 10.8; or, 10.8 lb. of ice is melted per pound of coal burned. Ans.

ADIABATIC-EXPANSION REFRIGERATION

FREE-AIR REFRIGERATING MACHINES

9. Air being the cheapest and most readily obtained gas, is for that reason used to some extent for the production of artificial cold. The machines in which it is used are known as **air-refrigerating machines**.

They utilize the fall of temperature that occurs when compressed air expands adiabatically and performs work, as stated in Art. 5.

The general arrangement of an air-refrigerating machine is shown in Fig. 1. The machine consists essentially of a compression cylinder A, an expansion cylinder B, a condenser R, and a cooler, or refrigerator box, D. The piston of the cylinder A is provided with suction valves V, V opening inwards, a discharge valve V', and a water-jacket J. The diameter of the cylinder B is slightly less than that of A.

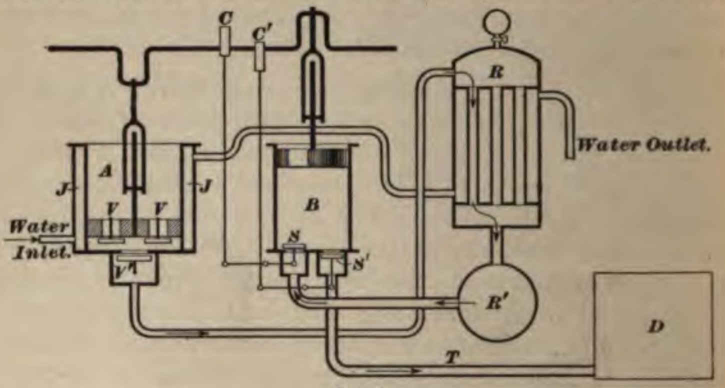


FIG. 1

The piston is solid, but the cylinder head is provided with two valves, an inlet valve S and an outlet valve S', which are operated by the eccentrics C and C'. The pistons are connected to cranks set at 180°. The condenser R is a surface condenser and receives a current of cold water from the water-jacket J of the compression cylinder A. A receiver R' is connected with the condenser and also communicates with the inlet valve S of the expansion cylinder B.

The air at ordinary pressure is taken into the cylinder A through the valves V, V, and is compressed adiabatically until the pressure becomes sufficient to open the valve V'. The air then passes into the condenser R, where it comes in contact with the cold surfaces of that vessel. The adiabatic compression has raised the temperature of the air; but in passing through the condenser, some of the heat contained

in the air is given up to the cold water circulating through the condenser, and the temperature is lowered nearly to that of the surrounding air. During this time, the valve S of the expansion cylinder B opens and permits an amount of air equal in weight to that expelled from A to pass from the receiver R' into the cylinder. The valve S closes and the air in the cylinder B expands, forcing the piston forwards and doing a certain amount of work. This expansion of the air in the cylinder B and the performance of work in forcing the piston forwards is at the expense of the energy stored in the air. The air therefore gives up sufficient heat to do the mechanical work, and as a result its temperature falls. As the air on entering B was at a normal temperature, the expansion brings the temperature below that of the surrounding objects. In other words, the air is cooled.

When the piston in B reaches the upper limit of its stroke, the valve S' opens; and as the piston descends, the cooled air escapes by means of the pipe T into the refrigerator box D.

The difference between the work done on the air in the compression cylinder and that done by the air in the expansion cylinder, and, in addition, the work required to overcome the friction of the entire machine, must be supplied by a steam engine or other motor.

10. Air at any ordinary temperature can hold a certain amount of water vapor in suspension. The limit, or point of saturation, that is, the point at which the air can hold no more water vapor, is called the **dew point.** When this point is reached, the excess of moisture above that which the air is able to hold is precipitated in the form of dew. The weight of moisture contained in a given volume of air at the dew point is not the same for all temperatures; in fact, air will hold in suspension four times the weight of moisture at 72° that it will at 32°. Assume that the air on entering the expansion cylinder B, Fig. 1, is at a temperature of 72° and is saturated with moisture. As the temperature falls during expansion, the water is gradually precipitated and condenses

on the walls of the cylinder. This water cools as the expansion goes on until it reaches 32°, when it freezes. The condensation, cooling, and freezing of the water greatly decrease the useful effect of the machine. Besides, the snow, which is the result of freezing the moisture, often gives trouble by clogging the valves.

11. The Haslam Foundry and Engineering Company, Derby, England, makes what is known as a *dry-air system*. They place a *drier* in the suction pipe from the condenser to the expansion cylinder. The compression cylinder *A*, Fig. 2,

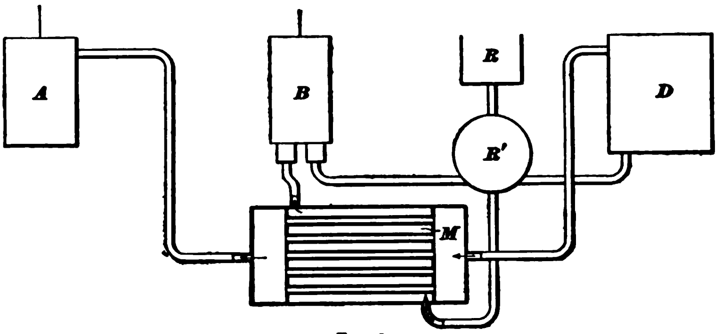


FIG. 2

takes the cold air from the refrigerator box *D*; on its way to *A*, this cold air passes through the pipes of the drier *M*. The cold strikes through these pipes and cools the air surrounding the pipes on its way from the receiver *R* to the expansion cylinder *B*. The air gives up a large percentage of its moisture in the drier, and the frosting in the cylinder *B* is much diminished.

ALLEN DENSE-AIR REFRIGERATING MACHINE

12. The air-refrigerating machines described in Arts. **9** and **11** take in air from the surrounding atmosphere at every stroke; in the **Allen dense-air machine,** however, air is used under an initial pressure of about 60 pounds per square inch and the same air is used over and over again.

A small supplementary pump attached to the machine charges the system and machine with air at the given

pressure, and serves to make up any loss due to leakage. The advantage of using air under pressure is as follows: The amount of heat that a given volume of air can abstract from a warmer body varies directly as the weight of the given volume. That is, if a given volume of air weighs 5 pounds, it can abstract five times the quantity of heat from the warmer body that can be abstracted by an equal volume weighing only 1 pound. This allows a smaller conveying pipe to be used, and also allows the machine to be placed at some distance from the refrigerating box or ice-making tank. Naturally, the smaller the pipe conveying the cold air to the refrigerating box, the less surface there is for the absorption of heat from the surrounding air and consequent warming of the cold air; hence, with the small pipe used in this machine, the cold air can be conveyed farther for a given rise of temperature.

13. A general perspective view of one form of the Allen dense-air machine is given in Fig. 3, and a diagrammatic

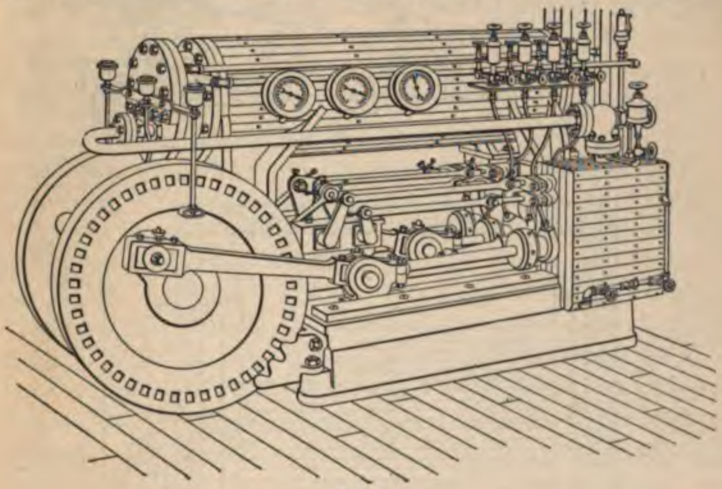
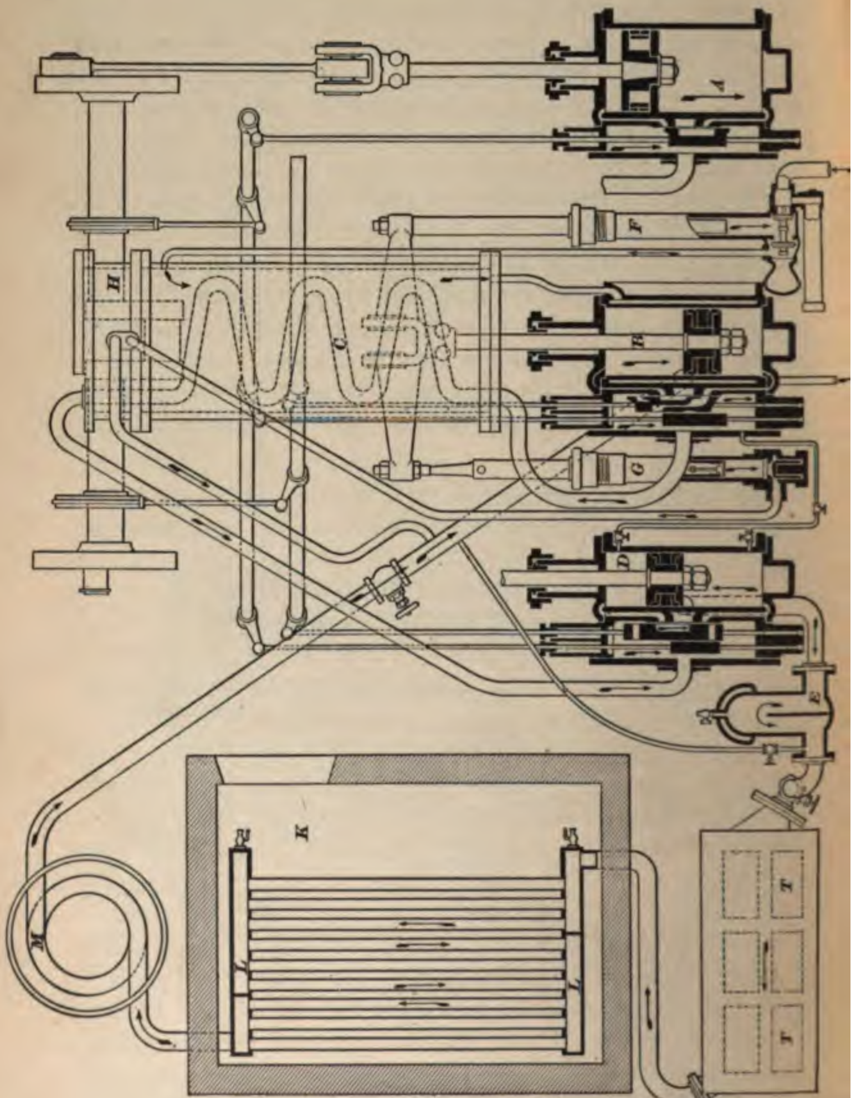


FIG. 3

illustration of a plant employing that machine is shown in Fig. 4. The machine consists of the following parts: a steam cylinder *A*, an air compressor *B*, an air expander *D*, a cooler *C*,


A
F
H
C
B
G
D
E
K
M
T
T
T

a water pump *F*, which supplies the cooling water to the cooler, a primer pump *G* for charging the machine with compressed air, and a trap *H*, in which the initial charge of air parts with nearly all of its moisture. The steam piston, air-compressor piston, and expander piston are coupled to the same crank-shaft.

14. The operation of the machine is as follows: On starting up, the primer pump charges the system with air taken from the surrounding atmosphere, compressing it to a pressure of from 60 to 65 pounds. This air, heated by the compression, is discharged into the trap *H*, where it is cooled by coming in contact with the cold head of the cooler *C*. On cooling, it deposits most of its moisture in this trap. The primer pump runs continually, and thus keeps up the initial pressure; any excess of air beyond that required is discharged through a small safety valve. From the trap, the compressed and cooled air passes into the double-acting air compressor *B*, where the initial charge is compressed to a pressure of from 210 to 225 pounds. The heat is abstracted from the air by passing it through a copper coil inside the cooler *C*, through which the cooling water is constantly circulated by the pump *F*. The air under high pressure is here cooled to nearly the temperature of the cooling water, and passes from the cooler to the expander cylinder, where it is cut off at one-third of the stroke, and in expanding does work. Its temperature is thus lowered to from 35° to 55° F. below zero. The expanded cold air, which is now at a pressure of from 60 to 65 pounds, then passes into an oil trap *E*, where it parts with most, if not all, of the lubricating oil used in the compressor and expander. All snow, due to unremoved moisture in the air, is gathered here. This trap is steam-jacketed; the steam is turned on, however, only when it is desired to move the oil and the snow from the trap. The air now passes through coils of pipe in the ice box *T* and refrigerating room *K*; the coils of the refrigerating room are shown at *L*. From there it passes through the drinking-water butt *M* and returns to the compressor inlet. By

passing the cold air through coils, a large heat-absorbing surface is provided; in its passage through these coils, the cold air absorbs the heat of the warmer air surrounding the coils, and thus cools it. Naturally, the cold air becomes warmer.

In some instances, the return air, which is still quite cold, is passed through a special cooler, where it cools the highly compressed air coming from the cooler *C*, thus furnishing the expander cylinder with cooler air than could be obtained otherwise. If this is done, the builder claims that a temperature of from 70° to 90° F. below zero is obtained.

EXAMPLES FOR PRACTICE

1. A refrigerating machine produces the refrigerating effect of the melting of 15 tons of ice in 24 hours; how many British thermal units does it abstract? Ans. 4,320,000 B. T. U.

2. A refrigerating machine producing the refrigerating effect of the melting of 20 tons of ice requires 5,000 pounds of coal in 24 hours to operate it; how many pounds of ice, in ice-melting effect, is this equivalent to, per pound of coal? Ans. 8 lb.

3. In example 2, if the coal cost $5 per ton, what will be the cost, per ton of ice, in the ice-melting effect produced? Ans. $.625 per T.

4. The ice-making capacity of the refrigerating machine mentioned in examples 2 and 3 is 60 per cent. of its refrigerating capacity; what will be the cost per ton of the ice made by it? Ans. $1.04 per T.

LATENT-HEAT REFRIGERATION

REFRIGERATING AGENTS

15. Requirements.— The air-refrigerating machine produces its refrigerating effect by means of the fall of temperature incident to adiabatic expansion. In all other refrigerating machines, the abstraction of heat is brought about by the vaporization of some liquid having a low boiling point. Such machines may be classed as **latent-heat refrigerating machines.** Theoretically, any volatile liquid may be used as a working fluid in a latent-heat machine; there are, however, various considerations of a

practical nature that govern the choice of the liquid. The chief requisites of the fluid used are: (1) It should vaporize at a low temperature when at ordinary atmospheric pressure; (2) it should have a high latent heat. The fluids that have been used in compression machines are *ether*, *sulphur dioxide*, *Pictet fluid*, *carbon dioxide*, and *anhydrous ammonia*.

16. Table I gives the boiling points, latent heats, and specific heats of various liquids at atmospheric pressure, 14.7 pounds per square inch.

TABLE I

PROPERTIES OF REFRIGERATING AGENTS

Liquid	Temperature of Boiling Point Degrees Fahrenheit	Latent Heat B. T. U.	Specific Heat of Liquid
Nitric acid	248.0		
Saturated brine	226.0		
Water	212.0	966.0	1.0000
Alcohol	173.0		
Chloroform	140.0		
Ether, Sulphurous	95.0	170.0	.5299
Ether, Methyl	−10.0		
Sulphur dioxide	14.0	168.7	.4100
Anhydrous ammonia	−28.5	573.0	1.0058
Carbon dioxide	−140.0	121.0	.9950

17. Ether.—Early in the history of ice-making and refrigerating machines, **ether** was almost universally used on account of its high condensing temperature and consequent low condensing pressure. This low condensing pressure made it possible to use compression pumps of ordinary construction, very much after the style and pattern of air pumps. However, the disadvantages of the use of ether were found to be very great; thus, the first cost of ether is considerable, it is very inflammable, and liable to explode when mixed with

air. Furthermore, owing to the density of the vapor at the required working pressure, the compression cylinder must be very large, viz., six times larger than for sulphur dioxide and seventeen times larger than for ammonia.

18. Sulphur Dioxide.—The objections to ether led to further investigation. **Sulphur dioxide** was found to be more efficient than ether, for though it required a higher condensing pressure, it did not require to be evaporated under a vacuum. Consequently, the compression pumps were made somewhat smaller for a given capacity, but were built stronger and more attention was given to the elimination of clearance spaces. The temperatures produced with sulphur dioxide, though lower than those obtained with ether, were not sufficiently low.

19. Pictet Fluid.—It was found by Professor Pictet, a Swiss physicist, that a mixture of 97 per cent. of sulphur dioxide and 3 per cent. carbon dioxide, commonly known as carbonic-acid gas, gives a boiling point 14° F. lower than pure sulphur dioxide. This liquid has been since known as **Pictet fluid.** Its latent heat has never been closely determined, but is probably nearly the same as that of pure sulphur dioxide.

20. Carbon Dioxide.—The lowest boiling point of any of the liquids employed at present in refrigeration is possessed by **carbon dioxide.** Under a gauge pressure of 200 pounds per square inch, it will boil at a temperature of about – 22° F. Its condensing pressure is correspondingly high, being about 900 pounds per square inch for a water temperature of 70° F.

21. Ammonia.—One atom of nitrogen combines with three atoms of hydrogen to form one molecule of **ammonia;** this is the only combination of these two elements. The ordinary ammonia of commerce is a solution of ammonia gas in water, and is properly known as *aqua ammonia.* The gas that passes off from the aqua ammonia is the ammonia formed by the combination of nitrogen and hydrogen. When

this gas is entirely free from vapor of water it is called **anhydrous-ammonia gas.**

Ammonia gas, when liquefied under a high pressure and allowed to evaporate under atmospheric pressure, gives a temperature of 28.5° F. below zero. Liquid anhydrous ammonia, when subjected to a temperature of − 115° F., freezes and forms a solid; in this state, it is almost odorless and is heavier than the liquid for a given volume.

Ammonia has no effect on either iron or steel, but rapidly corrodes copper and brass. It is therefore necessary to make the parts of ammonia machines out of the former metals. At a temperature of 900° F. the gas is resolved into its constituent elements. But is is probable that this dissociation occurs to a limited degree at much lower temperatures.

Ammonia is not inflammable at ordinary temperatures. The liquid will not explode, but when run into drums or flasks, room should be left for expansion. Like almost all liquids, ammonia expands when heated, and if sufficient space is not left the flask is likely to burst if exposed to a high temperature.

22. Aqua ammonia, known also as ammonia liquor, is a solution of ammonia gas in water. At 32° F. and under atmospheric pressure, water will absorb 1,140 times its volume of ammonia gas. The amount of gas held in solution affects the specific gravity of the solution; the more gas absorbed, the less is the density. The amount of ammonia that can be absorbed by water is governed by the temperature of the water and the pressure of the gas. The colder the water and greater the pressure, the greater is the quantity of ammonia taken up.

The strength of a solution of ammonia gas in water is measured by a hydrometer. When this instrument is placed in a liquid, it is evident that it will sink deeper the less is the density of the liquid; hence, the density will be indicated by the mark on the scale at the level of the liquid. For liquids lighter than water, the point to which the instrument sinks when placed in a solution of ten parts of salt to ninety of

water is marked 0°, and the point to which it sinks in distilled water is marked 10°. The space between the two marks is divided into ten parts and the division is continued to the top of the stem. The hydrometer thus graduated is generally used for ammonia solutions, though there is another graduation in which the reading for pure water is 0° instead of 10°.

TABLE II

STRENGTH OF AMMONIA LIQUOR

Percentage of Ammonia by Weight	Specific Gravity	Degrees on Hydrometer	
		Water 10°	Water 0°
0	1.000	10.0	0.0
1	.993	11.0	1.0
2	.986	12.0	2.0
4	.979	13.0	3.0
6	.972	14.0	4.0
8	.966	15.0	5.0
10	.960	16.0	6.0
12	.953	17.1	7.0
14	.945	18.3	8.2
16	.938	19.5	9.2
18	.931	20.7	10.3
20	.925	21.7	11.2
22	.919	22.8	12.3
24	.913	23.9	13.2
26	.907	24.8	14.3
28	.902	25.7	15.2
30	.897	26.6	16.2
32	.892	27.5	17.3
34	.888	28.4	18.2
36	.884	29.3	19.1
38	.880	30.2	20.0

The specific gravity of aqua ammonia and the percentage of ammonia gas, corresponding to a given hydrometer

reading on either of the graduations mentioned, are given in Table II. In this table, the first column gives the number of parts of ammonia gas in one hundred parts of the solution; the second column gives the specific gravity of the solution; and the third column gives the corresponding reading on the hydrometer. For example, if the hydrometer reading is 16°, the solution consists of ten parts, by weight, of ammonia to ninety parts of water, and the specific gravity of the solution is .960, the point to which the hydrometer sinks in distilled water being marked 10°.

23. All chemical actions, as well as solutions and absorptions, are accompanied by an increase or decrease in the temperature of the mixture. This is especially true of absorptions. In the case of ammonia absorbed in water, 925.7 British thermal units is given up for each pound of ammonia gas absorbed under atmospheric pressure. Though no very exhaustive experiments have been made on this subject, results deduced from the practical running of refrigerating machines show that this figure is practically constant. Since heat is given up when ammonia gas is absorbed, heat will be absorbed when the gas is again liberated from the water. The quantity of heat necessary to liberate 1 pound of anhydrous gas is 925.7 British thermal units, the same amount that is given out by the liquid when the gas is being absorbed.

24. If it is desired to test the purity of liquid anhydrous ammonia, draw some into a flask having a cork with a bent tube inserted in it. Wrap the flask in dry waste or cloth before drawing off the ammonia, or the fingers may be frozen fast to the flask. The liquid ammonia evaporates slowly, the gas passing out of the bent tube. If an accurate low-temperature thermometer is immersed in the boiling liquid, it should indicate a temperature of $-28.5°$ F., with normal barometric pressure. If the liquid is pure anhydrous ammonia, there should be no residue left in the flask; a deposit of oil or water indicates impure ammonia.

To detect a leak in piping, in case the odor does not betray it, hold a glass rod moistened with muriatic acid near

the supposed leak; a white fume rising from the rod indicates an escape of ammonia.

To detect ammonia leaks in piping under water or brine, add to a sample of the suspected liquid a few drops of *Nessler's reagent;* a yellow coloring indicates traces of ammonia, but if the quantity of ammonia is large, the color changes to a dark brown. To prepare Nessler's reagent, dissolve $\frac{3}{8}$ ounce of mercuric chloride in about $10\frac{1}{2}$ ounces of distilled water; dissolve $1\frac{1}{4}$ ounces of potassium iodide in $3\frac{1}{2}$ ounces of water; add the former solution to the latter, with constant stirring, until a slight permanent red precipitate is produced. Next, dissolve $4\frac{1}{4}$ ounces of potassium hydrate in about 7 ounces of water; allow the solution to cool; add it to the above solution, and make up with water to $35\frac{1}{4}$ ounces, then add mercuric-chloride solution until a permanent precipitate again forms; allow it to stand until settled, and decant off the clear solution for use; keep it in glass-stoppered blue bottles, and set away in a dark place to keep it from decomposing.

25. Cooling Effects of Various Refrigerating Agents.—The density of the gas at the evaporating temperature and the latent heat of the liquid determine the size of the compression cylinder necessary for any required capacity. The same machine working between 5° and 64.4° will give the following cooling effects per cubic foot of compressor-piston displacement under theoretically perfect conditions:

	British Thermal Units
Carbon dioxide	248.18
Ammonia	62.75
Sulphur dioxide	22.88
Sulphuric ether	3.68

AMMONIA-COMPRESSION SYSTEM

26. General Description.—Suppose that a flask or ordinary bottle *B*, Fig. 5, supplied with a cork having a bent tube *G* inserted, is partly filled with anhydrous ammonia.

This can be done easily, as the evaporation of the ammonia is comparatively slow, owing to its high latent heat. As the ammonia enters the flask, frost will begin to gather on the outside. If this flask is now placed in a pail *A* partly filled with water *C*, in a short time ice *D* will begin to gather on the outside of the flask. This is the simplest form of ice machine, but in this form the liquid ammonia, when it evaporates, passes out of the flask and is lost.

As with all volatile vapors, the temperature at which vaporization (or condensation) occurs rises as the pressure of the vapor increases. To prove this, insert a thermometer into the flask so that the bulb is immersed in the boiling ammonia. The temperature will fall rapidly, and, if the thermometer is correct, should register 28.5° below zero. Take a piece of pipe; weld or plug one end and fit the other with a cap *B*, Fig. 6. Arrange a stuffingbox *C* about the thermometer *T* in the cap. Also provide an opening connecting with the pressure gauge *P*. Unscrew the cap and pour the contents of the flask into the pipe *A*, and screw on the cap *B*. If the gauge points to zero, the thermometer should still read −28.5° F. Watch the gauge and thermometer carefully. The ammonia evaporating in the pipe liberates gas. As this gas cannot escape, it creates a pressure in the pipe, which will be shown on the gauge, and a corresponding increase in the temperature of the boiling ammonia will become apparent. This will continue until the temperature of the liquid will be identical with the surrounding objects. Assume this temperature to be about 70° F.; the gauge should then show a pressure of 130 pounds per square inch. If, therefore, the temperature of the pipe is kept at 70° by immersing it in water at that temperature, and a pressure slightly in excess of 130 pounds per square inch is kept in the pipe, no further evaporation will take place, and the remaining liquid ammonia will lie quietly in the pipe.

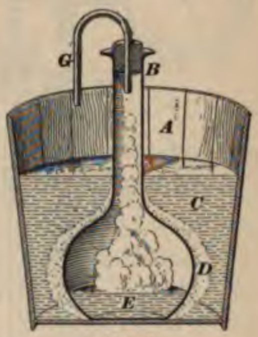


FIG. 5

27. It is apparent that if some means be devised of taking the evaporating gas as it leaves the flask in Fig. 5 and transferring it into the pipe of Fig. 6, it will be possible to save the gas. In place of a short piece of pipe *A*, Fig. 6, submerge a large coil of pipe *A*, Fig. 7, in a tank of water *C*. The water enters by means of the pipe *F*, and overflows by the pipe *F'*; the continuous flow tends to keep the temperature of the coil *A* constant. Replace the flask *B*, Fig. 5, with a coil of pipe *B*, Fig. 7, immersed in a water tank *D*. Provide a pump capable of working against a high pressure, and connect the suction of the pump with the coil *B*, and the discharge with the coil *A*. Also provide pressure gauges *G'*, *G* on each line. Connect the bottom of the two coils together, and place a valve *E* in the line. Partly fill the coil *A* with anhydrous ammonia. If the temperature of the water in *C* is about 70°, the gauge *G'* will show a pressure of 130 pounds. Open the valve *E* slightly, and leave it open. The pressure denoted by the gauge *G* will gradually rise, and ice will begin to form on the lower pipes of *B*. When the pressure shown by *G* has reached 15 pounds, start the pump *P*, which will draw the gas out of the coil *B*, compress it, and deliver it to the coil *A*. The gas entering *A*, which has been heated by the compression, comes in contact with the cold pipe surface, and is first cooled until its temperature is but little above that of the condensing water flowing out through *F'*. The gas then condenses and falls to the bottom of the coil in the form of

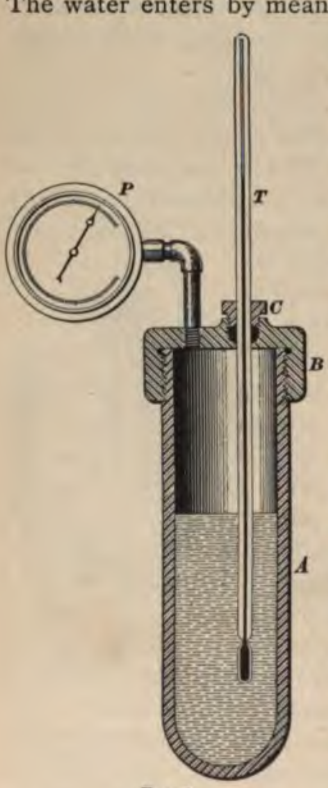


FIG. 6

liquid anhydrous ammonia. As the valve *E* is open, the coil *A* is prevented from filling up. The withdrawal of a quantity of the gas in the coil *B* tends to decrease the pressure in that coil; however, a quantity of the liquid passes from

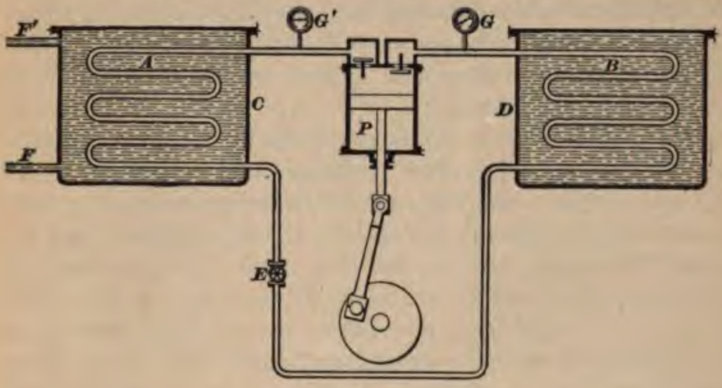


FIG. 7

A to *B*, through the expansion valve *E*, vaporizes, and supplies an amount of gas equal to that withdrawn by the pump.

28. Dry Compression, Wet Compression, and Oil Injection.—If ammonia vapor is compressed adiabatically, it will be superheated, as the work done on the vapor by the piston is stored in the vapor in the form of heat. This heat must be got rid of during the period of compression, since otherwise it must be absorbed by the condensing water before the vapor can be condensed. It is most economical to remove the heat, as far as possible, as fast as it is generated, and to keep the temperature of the cylinder comparatively low throughout the compression. In fact, it is absolutely necessary to employ some method of keeping the cylinder cool, otherwise the excessive heat developed in compression will soon become so great that the gas will enter the cylinder in a greatly superheated state, which will lessen its density. This decrease in density will naturally cause a corresponding decrease in the weight of gas pumped in a given time, thus affecting both capacity and economy.

29. Various expedients are resorted to for the purpose of abstracting the heat of compression. The simplest of these is jacketing the cylinder with water; this method is known as the **dry-compression system.** The gas enters the cylinder in a nearly saturated state; the instant that compression begins, the vapor would become immediately superheated if the heat were not carried off by the cold water surrounding the cylinder.

The majority of compressors built in the United States are of the water-jacketed, dry-compression type. In the case of vertical compressors, the water-jackets are merely small tanks enclosing the walls of the cylinder, and are sufficiently high, so that the top head of the cylinder is also immersed. They are open at the top and the water passes off by gravity. Horizontal compressors are usually water-jacketed on the cylinder walls only, the heads being unjacketed.

30. In the **wet-compression system,** the cylinder is not jacketed, but a certain amount of liquid anhydrous ammonia is allowed to enter the cylinder with each stroke of the compressor; the mixture of vapor and liquid remains saturated while it is compressed, since the heat equivalent of the work of compression is taken up by the vaporization of a part of the liquid, and the vapor remains at the temperature due to the pressure.

The injection of a small quantity of anhydrous ammonia to cool, by its evaporation, the walls of the cylinder was the invention of Prof. C. P. G. Linde, of Munich, Germany. The machines built under this system bear his name and are of the horizontal, double-acting type. The temperature of the gas leaving the compressor in case of the Linde machine is much lower than that in the dry-compression system, and the theoretical economy is somewhat higher.

31. A third method of removing the heat of compression is employed in the so-called **oil-injection system,** which is a modification of the wet-compression system, and is used by the De La Vergne Refrigerating-Machine Company, of

New York. Instead of permitting anhydrous ammonia to enter the cylinder, a certain quantity of oil is injected during the stroke; the oil cools the gas during compression, seals the valves, and cuts down the clearance space.

In the earlier machines, a small quantity of oil was admitted into the compression cylinder during suction and was expelled at compression. The mixture of oil and gas at a somewhat high temperature passed to the oil separator, where the oil was separated from the ammonia gas. The gas passed on to the condenser in its regular cycle; the oil was taken from the separator, passed through a cooling coil immersed in running water, and was then allowed to run into a receiver, from which it again passed into the suction pipe. As by this method a certain quantity of oil was allowed to enter with the gas, the volume of the gas entering the cylinder at each stroke was decreased in proportion to the amount of oil injected; in case a considerable quantity of oil was fed in, this would cut down the capacity appreciably. In order to obviate this difficulty, the De La Vergne Refrigerating-Machine Company, in its new compressors, injects oil by means of a small pump after the work of compression has set in, and not during suction as formerly. This also permits the oil to be kept fully charged with ammonia.

AMMONIA-ABSORPTION SYSTEM

32. The action of a refrigerating system of the absorption type is based on the affinity of a vapor, usually ammonia vapor, for water. If heat be applied to a strong aqua-ammonia solution, the ammonia gas or vapor will be driven off at a relatively high pressure, and when passed through a condensing coil will condense to the liquid state. The liquid can now, just as in the compression system, be admitted to a refrigerating coil through an expansion valve. In this coil, it will vaporize and withdraw heat from the surrounding objects. The vapor may now be again absorbed by water, thus regaining its original state and closing the cycle of operations.

33. The essential features of the absorption system are shown in Fig. 8. Steam is admitted, at a gauge pressure of about 40 pounds per square inch, to a coil B submerged in a strong solution of aqua ammonia contained in the vessel A.

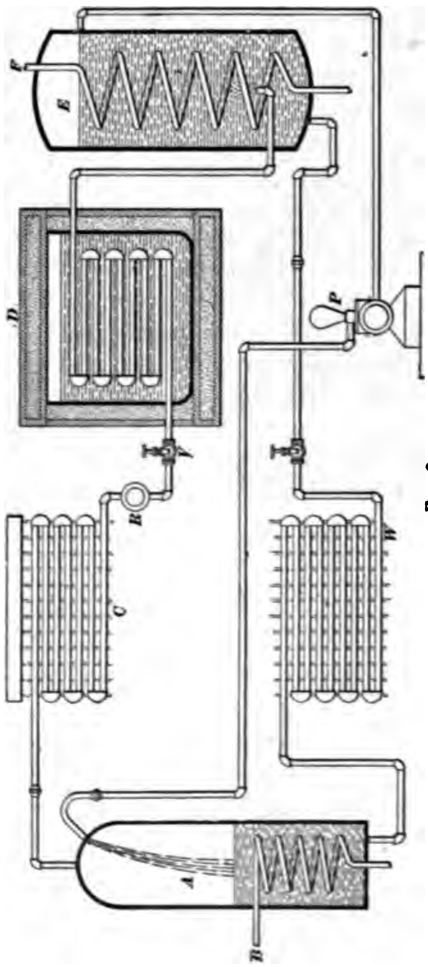


FIG. 8

The temperature of the solution will be raised nearly to that of the incoming steam, say to about 270° F., and the heat absorbed will cause the ammonia gas to be driven off at a pressure of, say, 160 pounds per square inch.

As the temperature of the solution is below the boiling point of water for this pressure, no water will evaporate, and only ammonia gas will pass over into the condenser C. The cold water flowing over the condensing coils absorbs heat from the gas, and the combined effect of the high pressure and the cooling action of the water liquefies the gas.

As the ammonia liquid passes through the expansion valve to the expansion coils, the pressure is reduced, reevaporation begins in the expansion coils, and heat is absorbed from the brine or other substance in the tank D.

In the compression system, the ammonia pump draws the gas from the expansion coils; but in the absorption system, the removal of the gas is effected by allowing the gas from the expansion coils to mingle with the weak solution of ammonia from which the gas was expelled in the still, or generator, *A*.

During the process of generating the ammonia gas in the still *A*, the strong solution rises to the top on account of its smaller specific gravity, and the weaker solution settles to the bottom and flows through a pipe to the vessel *E*, called the **absorber.** Here it meets the gas as it comes from the expansion coil and absorbs it. Since a low temperature is required for efficient absorption, the weak liquor on its way to the absorber passes through a coil *W* that is cooled by running water. The absorber *E* is also provided with a water coil *F*. A small pump *P* takes the strong liquid from near the top of the absorber and forces it back into the generator; this completes the cycle of operations.

APPLICATION OF REFRIGERATION

REFRIGERATING SYSTEMS

34. There are two widely used systems of refrigeration, viz., the *brine* and the *direct expansion*. In the former system, the expansion coils are immersed in a tank of brine; this brine, which is a non-freezing solution, gives up its heat to the ammonia evaporating in the coils or to the cold air passing through them; it is then pumped through coils of pipe placed on the sides or ceiling of the room to be cooled. The circulating brine thus continually absorbs heat from the cold room and gives it to the ammonia or air.

In the **direct-expansion ammonia,** or **air, system,** the ammonia or air is admitted directly into the coils in the rooms to be refrigerated. The heat of the cold room is taken up by the ammonia or air directly, and the intermediate agent, brine, is not employed. The difference between the two systems may be explained as follows: In Fig. 7,

suppose the expansion coil B to be a comparatively short or compact coil, and let the vessel D be a tank containing brine; this arrangement constitutes the brine system. On the other hand suppose the vessel D to represent the room or rooms to be cooled, and suppose the expansion coil B to be a long coil divided into many branches and located on the ceilings or sides of the rooms; this arrangement constitutes a direct-expansion system.

In another direct-expansion air system, the cold air is led through suitable chutes to the rooms to be cooled, and is discharged directly into the room. The air compressor draws its air supply from these rooms.

The proper temperatures for the refrigerating rooms depend on the nature of the article to be preserved. They are about as follows: fish, meat, and poultry, 20° F.; fruit and vegetables, about 35° F.

BRINE

35. There are two salts in common use for making the **brine** used in brine circulation. The first is Liverpool salt (chloride of sodium), which forms the ordinary brine capable of withstanding a temperature of about 0° F. This salt is cheap in first cost, but has a corrosive action on iron.

The other salt is the chloride of calcium. It has no corrosive action on iron, which makes it unnecessary to have the brine pump lined with brass. It is oily in nature, and for that reason has a strong tendency to leak if the piping is in any way imperfect; care should therefore be exercised in the pipework of a chloride-of-calcium brine circulation. It is possible to obtain much lower temperatures by the use of chloride of calcium than with chloride of sodium.

The cost of chloride of calcium is about double that of salt. The quality is extremely variable; insist on having *fused* chloride of calcium. The salt is excessively deliquescent, that is, it is capable of absorbing a large quantity of water; for this reason it is often used as a drier. This great avidity of calcium chloride for water renders adulteration by

the absorption of water very easy. Even the fused salt contains as much as 20 per cent. of water, whereas the unfused salt, though still in solid form, contains upwards of 50 per cent. of water. Care should, therefore, be used in selecting the salt.

When it is desired to purchase chloride of calcium, request samples. Dissolve a certain weight of each sample of the salt in the same quantity of water; take a hydrometer reading of each of the samples after the salt is thoroughly dissolved; the one giving the highest reading is the best sample.

To make chloride-of-calcium brine, use about equal weights of water and chloride of calcium. Whenever it is observed that the brine commences to freeze, add a little chloride of calcium, stirring it well into the brine.

ICE MAKING

36. There are now two systems in use for making ice, viz., the *can system* and the *plate system*. The can system is the more common of the two, being cheaper in first cost and requiring less attention in manipulation. The plate system, however, has the advantage of giving a clearer ice.

The apparatus used in the **can system** consists of a large rectangular wood or iron tank containing the expansion coils or pipes. Galvanized-iron cans are placed between the rows of expansion coils. These cans are filled with distilled water, and when the brine is chilled below the freezing point of distilled water, the water in the cans freezes. If the temperature of the brine is not allowed to fall below 25° and ordinary well water is used in the cans, the ice produced will be comparatively clear on the outside and rather snowy in the center. If, however, the brine temperature is allowed to fall to about 15°, the ice will be entirely opaque.

In the **plate system**, the refrigerating fluid is circulated through vertical cast-iron or wrought-iron hollow plates, set on edge in a tank of water. Ice begins to form on the plates and gradually extends out into the tank. If the temperature of the plates is kept comparatively high at the start until 2 or 3

inches of ice is formed, and then gradually reduced as the ice formation increases, a clear cake or plate of ice will be formed on each side of the plates. The refrigerating fluid is then drained from the plates, and warm water is introduced, which thaws the ice adhering to the sides. As soon as the ice is detached, it floats to the surface of the water in the tank.

RUNNING REFRIGERATING MACHINES

RUNNING AN ALLEN DENSE-AIR MACHINE

37. Warm up the steam engine. Open the suction valve and discharge valve of the circulating pump. See that the two valves in the main pipes are open, and that the valve of the by-pass pipe connecting the return-air pipe with the cold-air pipe is closed. Open the blow valves of the expander cylinder and the petcocks of the traps. Start the machine; when no more grease or water discharges from the various petcocks, close them. Be sure that the circulating water is in motion.

During running, open the petcocks of the water trap *H* frequently enough to allow the water collected there not to fill it more than half full. Once or twice a day the machine should be cleaned by heating it and blowing out all oil and deposits; this is done as follows: First open the valve in the by-pass pipe. Then close the two valves of the main pipes. Open the valves in the hot-air pipe leading from the valve chest of the compressor to the expander cylinder and partly close the valve of the expander inlet pipe. Turn the live steam on the jacket of the oil trap, opening the outlet of the jacket just enough to drain off the condensed steam. Run in this manner for about ½ hour, and duriug this time frequently open the blow-off valves of the oil trap and expander, until the trap and expander are clear. Then shut off the steam from the jacket of the trap, drain the connections, close the valves in the hot-air pipe leading from the compressor to the expander, close all petcocks, open the

valves of the main pipes, and close the by-pass valve. The machine will now generate cold as usual. When a pressure of from 60 to 65 pounds cannot be retained in the system, it indicates a leakage of air somewhere. It is usually found at the stuffingboxes, which should then be tightened or repacked. If the compressor does not keep up its usual pressure in relation to the initial pressure, it shows that there is either a leak from the high-pressure to the low-pressure part of the system or to the atmosphere, or it may be due to the cup-leather packing of the compressor and expander cylinder having given out. The failure to maintain the high pressure will result in a higher cold-air temperature. The cup leathers should be made of white-oak tanned leather well soaked in castor oil. They will last from 1 to 2 months in steady use. The sight-feed lubricators for the compressor and expander cylinders should be filled with a light, pure mineral machine oil from which all paraffin has been removed. Use about three drops of oil per minute in the compressor and two drops in the expander. The makers of this machine recommend Leonard & Ellis extra machine oil for this purpose.

Whenever the pipes of the manifolds in the refrigerating room, ice tank, etc., are thawed out, drain them. Never omit to do this when the opportunity presents itself.

RUNNING AN AMMONIA-COMPRESSION MACHINE

38. The brine having been prepared, the machine charged with ammonia, and all connections tested, the apparatus is ready for service. Before any cooling is possible in the refrigerating room, it is necessary to cool the brine charge below the freezing point of water. The brine pump being stopped, the water pump is started and water is run over the condensers and through the water-jacket of the compressor. As the temperature of the brine is probably in the neighborhood of 55°, a comparatively high back pressure can be kept in the expansion coils. As the compressor is started when the pressure in the expansion coils is comparatively high

—100 to 150 pounds—it is necessary to reduce this pressure to 30 or 40 pounds before the brine shows any perceptible cooling. The main suction and discharge valves are opened, the by-pass valves are closed, and the compressor is started. The back-pressure gauge will begin to indicate less and less pressure. When a pressure of 30 pounds is reached, the expansion valve is slightly opened, and care is taken to regulate it so as to keep the back pressure between 30 and 35 pounds. After some time, the brine temperature will have fallen to about 25°.

The brine pump can now be started. This will start the circulation in the refrigerating room and also that in the brine tank, which will help to cool the brine more rapidly. The expansion valve can be closed a little, so that the back pressure will drop gradually as the temperature of the brine falls. The machine is now in full operation.

If at any time the compressor cylinder begins to groan, some oil should be pumped into the suction pipe. Care should also be taken that the piston-rod packing is well lubricated; if it begins to heat, the gland on the stuffingbox should be slackened and some oil pumped in. This will cool the rod. It is better to have the piston rod leak a little the first day or two than to have the packing so tight as to cut the rod.

39. When the brine has cooled to a temperature of 15°, an inspection of the charge should indicate at least 6 inches of liquid anhydrous ammonia in the receiver of the condenser and a back pressure of 20 to 25 pounds. If, however, the back pressure is lower and there is a small quantity of anhydrous ammonia in the receiver, a drum containing liquid anhydrous ammonia should be connected up and its contents pumped into the machine. There is usually one main expansion valve and a number of feed-valves, one on each coil, for the purpose of regulating the amount of anhydrous ammonia being fed to the separate coils. These valves should be adjusted so as to proportion the amount of anhydrous ammonia supplied to each coil; then they should be left alone. The total amount of anhydrous ammonia expanded should be

regulated by the main expansion valve. If the compressor is working properly, the whole action of the machine hinges on the proper manipulation of the expansion valve. A low back pressure is detrimental to the economical working of a compression machine. Therefore, care should be taken to carry as high a back pressure as possible, and at the same time avoid overfeeding. The best indication of overfeeding is a heavy frost on the suction pipe of the compressor. Not all the liquid anhydrous ammonia is evaporated in the expansion coils, and, consequently, some of it passes over to the compressor. The evaporation in the suction end of the cylinder causes the packing of the piston rod to freeze, and it is liable to leak as soon as it thaws out again. The greatest danger from overfeeding, however, arises from liquid ammonia or oil entering the cylinder of the compressor. The liquid being incompressible, its presence may result in the breakage of a cylinder head or of a shaft, or the derangement of some part of the compressor.

40. To shut down a compression machine, the main throttle valve of the steam engine is closed, care being taken that the compressor does not stop on a dead center. The valve on the oil feed is then closed, together with the main suction and delivery valves and the expansion valve; the other valves may be left open. The water pump is then stopped and all the drips are opened; the same is done with the brine pump. The refrigerating plant is now shut down, and the steam plant may be shut down as in ordinary practice.

RUNNING AN AMMONIA-ABSORPTION MACHINE

41. After the air and other gases have been expelled from the absorber, the ammonia pump is started, and the steam is gradually turned on the generator. When a pressure of 120 or 130 pounds has been reached, the valve leading to the condenser is opened and the water is turned on the condenser and absorber. This valve should be opened slightly, so as not to create too strong a current between the condenser and the generator, as the difference

between the pressures in these two vessels may be considerable. The generator pressure now extends to all the high-pressure parts of the machine. The expansion valve is then opened a very little until the gauge indicates 15 to 20 pounds. As soon as the liquor is seen in the gauge glass of the absorber, the suction valve and the delivery valve of the ammonia pump are opened and the ammonia pump is started. The pump is run at such a rate as to keep the liquor in the gauge glass on the absorber at a constant level. The machine is now doing regular work, cooling the brine in the brine tank.

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NOTE.—All items in this index refer first to the section and then to the page of the section. Thus, "Abaft, §9, p2," means that abaft will be found on page 2 of section 9.

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